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TESTING THE SYSTEMS OF THE AUTONOMOUS AGRICULTURAL ROBOT

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Abstract: The aim of the paper was to test of the concept of the navigation system for the autonomous robot for sowing and wide row planting. Autonomous work of the robot in the field of traction and agronomic processes is implemented based on data from many sensors (cameras, position sensors, distance sensors, and others). The robot is intended for ecologic cultivation requiring mechanical removal of weeds or in crops with application of selective liquid agrochemicals limited to the minimum. The use of a vision system, based on the map coordinates of the position of the sown seeds, allows for their care on an early stage of plant development. Main sensor system is based on a specialized GPS receiver and inertial navigation providing position information with an accuracy of around 10 mm. To determine the angular acceleration the IMU (Inertial Measurement Unit) is used. Additionally, information from the acceleration sensors and wheel encoders is used for navigation purposes. This system is used to: control the speed of the robot, keep the robot on the designated path, and detect the precise position of the seeds. The exact information of the seeds position is used to build maps of seeds, which will be used as supporting information for precision weeding, and to control the position of and operation of key components. The front camera view is used to increase positioning accuracy of the robot. It will allow corrections of the robot path regarding the rows of plants. The vision system is also used for detection of non-moving objects. A structure of requirements for the SQL database has been developed, which is used to store plant and weed geo-data, as well as store data about plants and weeds, based on images recorded by the vision system.

Keywords: AGRICULTURE ROBOT, CARE OF PLANTS, AUTONOMOUS WORK, POSITION OF PLANTS DATA BASE

1. Introduction

Syndicate of Industrial Institute of Agricultural Engineering in Poznań, with the Institute of Vehicles of Warsaw University of Technology and PROMAR company from Poznań started a design of autonomous farm robot for sowing and cultivation of wide row planting (Figures 1 and 2).

The aim of the project is to develop the construction and operation procedures for the autonomous robot for sowing and wide row planting. On the current stage of the product development laboratory and exploitation tests on an experimental model are carried out. Autonomous work of the robot in the field of traction and agronomic processes is implemented based on data from different sensors (cameras, positioning sensors, distance sensors, and others). Positive test results will allow for the use of the robot in organic crops requiring mechanical removal of weeds or in crops with application of selective liquid agrochemicals limited to the minimum. The use of a vision system, supported with the map coordinates of the position of the sown seeds, allow for their care on an early stage of plant development. The applicability of the robot to onerous work in organic farming may encourage farmers to discontinue the use of herbicides in crops.

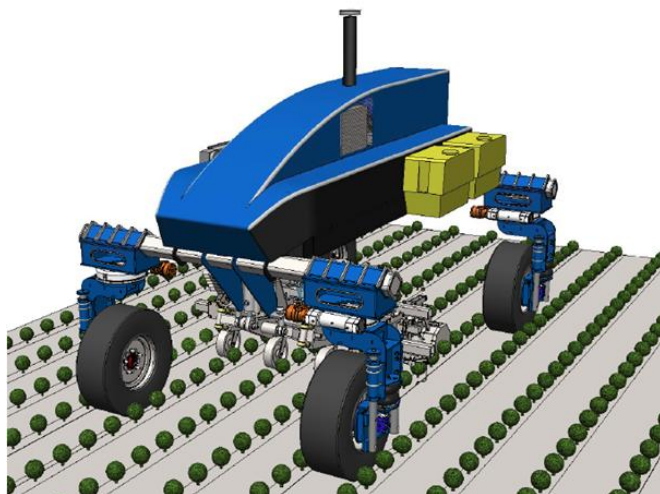


Fig. 1 CAD Model of the robot²

The autonomous farm robot is intended to work in different working conditions:

- terrain: empowered field, field roads,
- work 24 hours/day (for this special lightning system for cameras was implemented),
- work in areas with varying degrees of lighting and visibility,
- temperature: 5 to 40 °C,
- weather: average rainfall, moderate wind, fog,
- typical obstacles in the open area.

Projected robot enables complex care of field crops including: red beet, sugar beet, sweet corn, cabbage, lettuce, forest nurseries, orchard, production of vegetables and ornamental plants. Additionally, it should enable the mechanical destruction of weeds and, if necessary, precise application of crop protection formulations and fertilizers. Robot is equipped with interchangeable tool sets: seeders and smart weeders, and spraying tools, which uses digital image analysis for their control^{1,2}.

3. Concept of the navigation system

Main navigation system of the robot is based on a dedicated GPS receiver providing position information with an accuracy of around 10 mm. This system, in conjunction with the IMU (Inertial Measurement Unit) for determining the angular accelerations, is used to control the robot on the designated path and to provide precise position required during precision seeding, weeding and spraying. The exact information of the seeds position obtained during the process of sowing is used for creating maps of seeds, which is then used as a supporting information for precision weeding and spraying and control the position and operation of key components.

The front camera view is used to increase positioning accuracy of the robot. It allows corrections of the robot path regarding the rows of plants. The vision system is also used for detection of non-moving objects. Simultaneously, main vision unit is used for acquiring camera images immediately before active hoes and sprayer for detecting the exact position of growing plants. Additional information from the acceleration sensors and encoders built-in wheels is used for navigation purposes.

The navigation system enables:

- trajectory correction of the robot,
- precise work of active hoe,

- position adjustment and precise dosing of liquid fertilizer plant health products.

The exact position of the robot is obtained from the fusion of signals from precision GPS (in the test version standard GPS receiver Ublox NEO7) and integrated system of inertial and magnetic sensors VN-100³. VN-100 is a complete AHRS (attitude heading reference system) system integrating measurements from three axial sensors: acceleration, angular velocity and Earth magnetic field. All of the sensors have temperature compensated sensitivity and common values. Moreover, all the skewness of axes was calibrated. VN-100 device provides accurate information from all the sensors and estimates of the spatial orientation angles, DCM (direct cosine matrix) transformation matrix, and estimated values of linear accelerations and angular velocities in the absolute coordinates (NED – north/east/ down) independent from the angular orientation of the sensor. It is planned that two AHRS systems will be placed on one robot: first related to the vehicle and the second with a connection to the tools (e.g. seeder). AHRS integrated with the vehicle will provide the detailed momentary angular orientation and acceleration of the body while the second set allows better estimation of the momentary position of the tool thus allowing precise localisation of the seeds. Detailed information about their positions, stored in the internal database, will be used in further fieldwork related to the care of plants^{4,5}.



Fig. 2 Robot during sugar beet seeding work

4. Concept of the collision avoidance system

The collision avoidance problem is divided into two different subproblems: detecting the obstacle and bypass the obstacle^{6,7}.

The raw measurements from Laser scanner 2D are at first transformed into Cartesian coordinates. The obstacles are detected from the transformed measurements using clustering method. The whole clustering process is illustrated in Fig. 3. The initial positions for the clusters are gained from the set of known obstacles. The cluster initial position is added if known obstacle is in sight of the scanner.

The path tracking method is based on the Nonlinear Model Predictive Control. Vougioukas⁸ has used the Nonlinear Model Predictive Control (NMPC) method to control the position of the vehicle. Moreover, the collision avoidance was included into the controller by using additional cost from distance sensor readings. The controller was able to follow a predefined path as well as avoid collisions with static obstacles. The functionality was proven with simulations.

In the NMPC, the control values are calculated, so that the given cost function is minimized. The constraints of the optimization problem are obtained from the system model and the constraints of the states and control values. There are different ways to include the object avoidance into the NMPC. One way is to add additional constraints to the state values. Another way is to add an

additional cost from the obstacles or simply to modify the reference trajectory to go past the obstacle.

In this way, the modification of the cost function was chosen. The underlying path tracking cost function is not changed nor the reference trajectory, but the cost from state is modified. This is because of the calculation capacity and the possibility that the obstacles could move.

When the reference trajectory is near an obstacle, it cannot be followed without colliding to the obstacle. Therefore, it is irrelevant to keep the cost from the reference trajectory. Instead a cost that makes the vehicle drive past the obstacle should be added. The obstacle can be closer on the side of the vehicle.

The calculated distance to the edge of the avoided area is used in the cost function, when the obstacle is inside the avoided area or the obstacle is closer to the avoided area than the vehicle is to the original reference trajectory. the cost is calculated only from one obstacle. If there are multiple obstacles inside the avoided area, the one with the largest value of the distance from the obstacle to the edge of the avoided area is chosen. The same methods are also used for the cost from the trailer position.

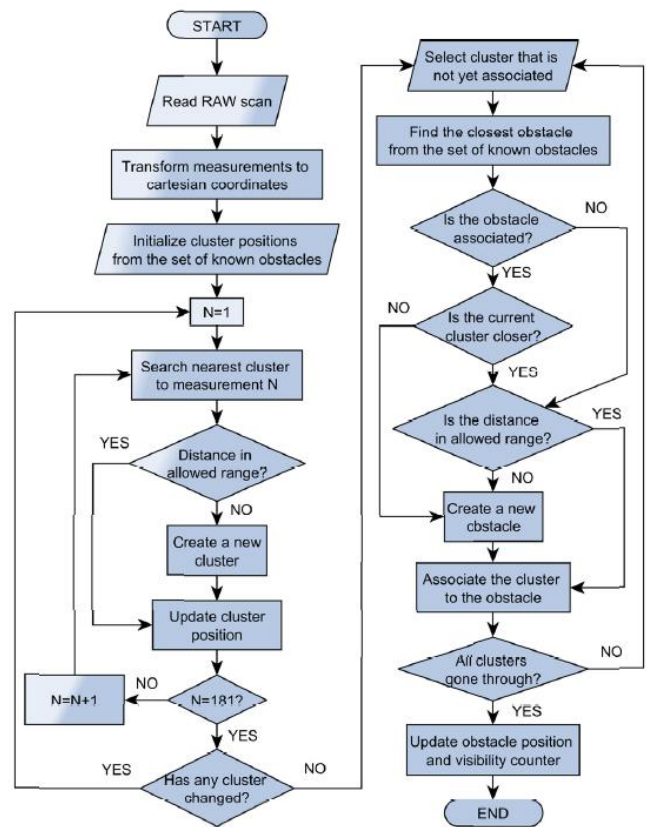


Fig. 3 The clustering algorithm⁸

5. Database system for storing information about the location and characteristics of crop plants and weeds

The plant and weed geo-data is stored in the specially developed SQL database. The database stores also data about plants and weeds, based on images recorded by the vision system.

The developed structure allows storing information about:

- geographical location of the plant (longitude and latitude parameters),
- date of measurement
- plant identifier in the database

The developed structure also allows storage of:

- graphic files representing photos of the analysed plants,
- plant features acquired through image processing and analysis.

During the preparation to the mission data from the SQL database are being transformed to the robot system using the JSON format⁹.

6 Testing of the vision system

The position of each plant must be determined for intra-row weeding. This means that plants must be classified into two classes, i.e., sugar beet (sweet corn) or weed. The pictures were taken with a normal colour photo camera and later digitised. For analysis of the object in the image it is essential to distinguish between the object of interest, here plants, and the background, here soil. In our case this means that we use the grey level distribution on the normalized green component. A total of seven shape features were selected: (area, perimeter, compactness, elongation, solidity, form factor and convexity) (Fig. 4).

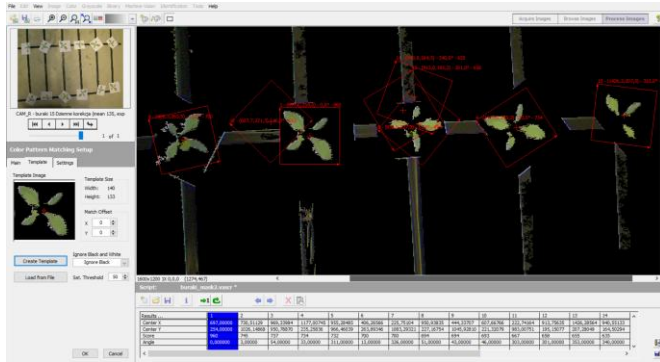


Fig. 4 Positions of plants identification

The first operation was to check the depth of field of the cameras mounted on the vehicle. The 8.5 mm lens provided theoretical sharpness in the range of 200 mm to infinity. During the tests, it turned out that the only reasonable choice was aperture 8.0, which reduced the depth of field. After properly calibrating the focus, using suitable markers, we managed to obtain a fairly sharp image for several plant height tests.

Another test task was to investigate the effect of lighting changes on the camera image. The tests were performed in full sun (Fig. 5) and in shadow (Fig. 6)¹⁰. This was the biggest problem for the vision system as this affects the quality of plant detection. This problem was solved by automatically changing the camera exposure time based on the analysis of grey level histograms.

Best results of plant detection were obtained by normalizing green component and the area. These features were used to classify sugar beets (green) and weeds (red)^{11, 12} (Fig. 7).

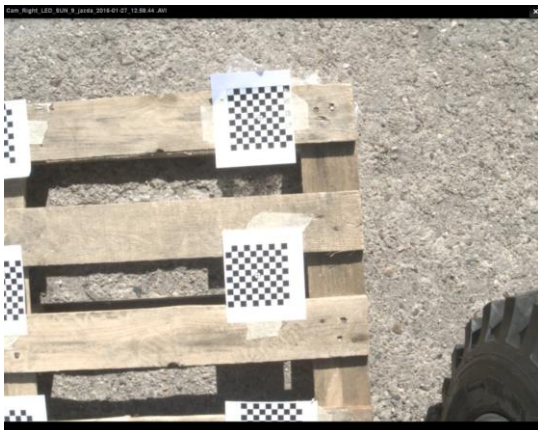


Fig. 5 Picture from the camera - lighting: the sun



Fig. 6 Picture from the camera - lighting: the shadow

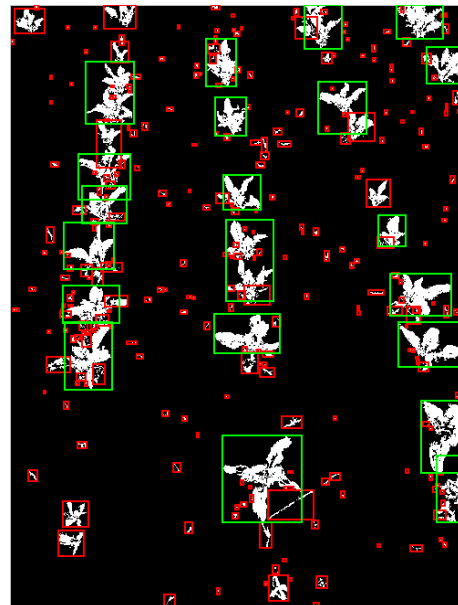


Fig. 7 Top - colour camera picture; Down - example of sugar beets (green), and weeds (red) classifications¹²

7. Mission tests

The control system of the robot is distributed (divided) into two controllers that communicate with each other through Ethernet network. The task of the navigation and path planning controller is to control robot Diesel engine and hydraulic systems and to drive the robot along the selected path. It also feeds the mission planning controller with the current position data necessary for the tools control.

The main tasks of the mission planning controller are: processing stage mission data sending them to the navigation controller, performing image analysis from cameras for sowing and weeding and controlling appropriate tools (eq. hoe or sprayer). Vision analysis performed on this controller is also used for obstacles avoidance.

The robot mission (e.g. sowing or weeding the whole field) is divided into the several primitive operations – stages: drive straight, drive to point, turn around, etc. Each of these operations begins in the current robot position and is described by the robot ending position (waypoint), velocity during drive and robot direction to be obtained in the end of the stage. Dividing the whole task into primitive operations significantly simplifies the algorithms of machine control as the robot control algorithm should be able to perform just a finite set of single, simple tasks. This also simplifies the process of testing the robot moving algorithms. For the task of testing the robot drive control algorithms the special robot simulator was developed allowing testing robot commands.

8. Conclusions

Field tests of the autonomous robot for sowing and wide row planting are currently under way. The tests of the navigation systems shown that the accuracy of the GPS position is as planned (~10 mm) however obtaining the proper fix and accuracy takes significant time: around 10 minutes. This also requires setting a base station in the field within the reach of the GPS radio communication.

The vision system for sugar beet/weed and sweet corn/weed classification was build and positively tested in laboratory conditions. Currently the field test on the growing plants are under way. The problem of camera's focus on the soil and the effect of the different lighting on the generated image was located and solved. Optional power of additional LED lighting and a method of sugar beets and weeds classification were also developed.

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THE EVALUATION QUALITY OF SOWING INDEPENDENCE WITH SELECTED PROPERTIES OF SEEDS

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Abstract: sowing quality means required vertical (depth) and horizontal (spatial) spacing of seeds in the soil with minimal damage to the seeds. The spatial distribution of seeds (horizontal spacing) is given by distance between the rows and spacing of the seeds in a row. The objective of the study was the evaluation and comparison of the sowing quality of sowing machine jd maxemerge xp with mechanical filling of ladle holes with holding fingers within usage of seeds with different dimensional and shape attributes in agricultural conditions. Sowing machine jd maxemerge xp reached good results for the quality especially when using the angular seeds. It reached the sowing quality expressed by the standard deviation of 46.95 - 50.76 % when seeds pr 37 n01 and for seeds pr 37 n01 it is 47.43 - 48.73 %. Another evaluative criterion is dual sowing of seeds or plants (drr), it was reached the average value 3.65 - 2.45 % for angular seeds and the average of 1.54 – 3.71 % for approximately spherical seeds.

KEY WORDS: SOWING QUALITY, ATTRIBUTES OF SEED, MAIZE,

INTRODUCTION

Efficiency of mechanized crop harvest is affected by the state of plants condition as well as whole range of other biological and technical factors. Therefore, the base for achievement of a generous harvest is achievement of high germination and uniform emergence of growth. In a common practise, farmers use seeds produced by a different manufacturers. These seeds are often dimensionally erratic and that affect, among other factors such as quality of soil preparation, the quality of sowing significantly (Gorzelay, 2009).

The spatial distribution of seeds (horizontal spacing) is given by distance between the rows and spacing of the seeds in a row. The distance between rows is adjustable within a sowing machine according to seeding crops. Final seedling emergence and the rate of speed of emergence are influenced by bulk density and aggregate size of seedbed (Dimitrov et al., 2012).

Precision sowing has been a major thrust of agricultural engineering research for many years; however, most of the research and development work has dealt with seeders for agronomic crops (Karayel et al., 2004). It was also stated that a seeders should place a seed in an environment in which the seed will reliably germinate and emerge. A number of factors affect the spacing of plants. The seed selection mechanism may fail to select or drop a seed resulting in large spacing between seeds. The mechanism may select and drop multiple seeds resulting in small spacings between seeds. Seed quality, soil conditions, seeder design and the skill of the operator all play a part in determining the final plant stand. Various types of cleaning, grading, separation and sowing equipment are designed on the basis of the physical properties of seeds. However, no model has been found to describe seeder parameters such as vacuum pressure related with physical properties of seeds.

Later on Karayel and Özmerzi (2002) stated that the variability in seed spacing with precision vacuum seeders increased with increasing forward speed. They found that a forward speed of 1 m.s⁻¹ consistently produced a better seed pattern than 1.5 and 2.0 m.s⁻¹ for precision sowing of melon and cucumber seeds. Field tests

showed that the nominal sowing depth of 60 mm was optimum according to sowing depth uniformity and emergence rate index.

In our case, we focused on sowing the maize (0.762 m). Since the distance between rows during sowing has not changed criterion for evaluation of accuracy of the horizontal spacing of seeds is the distance between seeds in a row (Findura et al., 2008). Seeds variability in the row is affected by soil preparation, physical attributes of seeds, parameters of sowing unit, working speed (Findura et al., 2008). According to (Páltik et al., 2002), following (a,b,c,d) technical parameters of the sowing-machine mostly affect the seeds' distance in row: a) fixing technique of the sowing unit on the frame of sowing machine (parallelogram); b) parameters of the sowing unit (shape of sowing foot, sowing speed within the seeds fall off sowing disc, etc.); c) type of sowing system; d) vertical acceleration of sowing unit and operating speed of the machine, etc (Streit et al., 1992).

The objective of the study was the evaluation and comparison of the sowing quality of sowing machine JD MaxEmerge XP (Fig.1) with mechanical filling of ladle holes with holding fingers within usage of seeds with different dimensional and shape attributes in agricultural conditions.

MATERIAL AND METHODS

Measurements were made on the farm Agrodivízia (Agro-division) Selice, Ltd. Process of soil cultivation produced the blackland soil type - mainly carbonated blackland, blackland "cernicka" and blackland's "arenicka". These soils are characterized as deep soils with deep humus horizon. They are clayey-sandy up to clayey soils with alternating graininess on the horizons of soil profile. Humus content in topsoil was determined at range 1.3 to 3 %. Soil reaction in topsoil was determined as pH 7.1 to 7.6. Soil samples were collected by soil measurement equipment NH-90 (Fig.2A). Measurement equipment was fixed on quadricycle Polaris 550 EFI X2. The samples were collected from soil depth 300 mm.

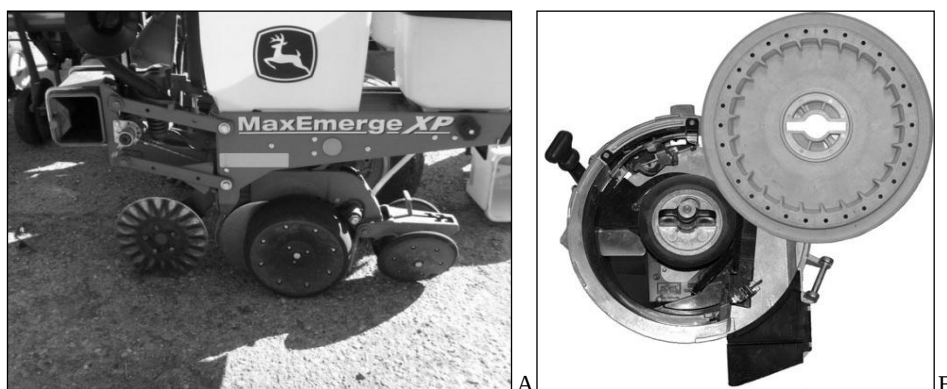


Fig.1: A - Sowing-machine JD MaxEmerge XP; B - Detail of sowing system

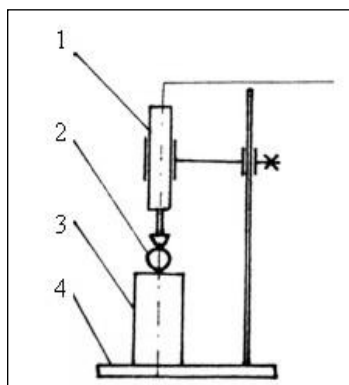


Fig.2: Technique of measurement of grains dimensional characteristics: 1- induction sensor, 2- grain, 3- rest, 4- laboratory metal holder

The sowing depth is also considered as one of the key factors. The best uniformity of sowing depth was obtained at the nominal sowing depth of 60 mm. For both seeders, either decreasing or increasing the nominal sowing depth from 60 mm resulted in a higher coefficient of variation for depth. The maximum emergence rate indices occurred when the nominal sowing depth was 40 and 60 mm, and the least mean emergence time was obtained at the nominal sowing depth of 40 mm. It can be concluded that the position of the seed in the soil effects mean emergence time and the emergence rate index of plants.

The sowing quality was evaluated within the sowing-machine JD MaxEmerge XP. This seeding machinery allows sowing of wide-rowed crops within the conventional as well as the soil conservation technologies. The sowing machine JD MaxEmerge XP operates on the principle of mechanical filling of ladle holes where seeds are drifted by holding fingers to the place of seeds falling. A variety of influences will affect the variability of seeds distances (later plants in the row) that will cause an inaccuracy of seeds placement to a larger or lower degree.

Apart from the impact of sowing system there will also operate other working parts of sowing unit within storage, covering and pushing the sow seeds into the sowing furrow. Unwanted seeds rolling and thus inaccuracy of seeds placement in the furrow also cause among the mentioned factors in introduction high of seeds falling (Table I).

Table I: Selected technical data of tested sowing-machine

Type of Sowing machine	JD MaxEmerge XP
High of seeds fallin,mm	505
Diameter od sowing disc, mm	370
Number of ladle elements	15
Fixing of sowing unit	parallelogram
Dept conduction of sowing foot	supporting wheel
Type of sowing foot	double disc

The procedures of shape and dimensional aspects of seeds, methodology of measurements and used devices in this study was obtained and adopted according to Findura at al. (2008). Shape and dimensional attributes of used seeds affect the reliable ladling of seeds and therefore also reaching symmetrical distances in the row. Device iM1610 and converter with amplifier were used for measuring the dimensional characteristics of grain. Our goal was to digitize the process of measurement. A/D converter MA-UNI with the amplifier and the measuring device with data-logger served as the base again. Sensor type W50 with high measurement accuracy (0.2 %), in this case, was used from the company HBM (Hottinger Baldwin Messtechnik). It is the sensor that works on the induction principle of measuring position. Data were automatically recorded into a data file individually in the order: depth, width and length after calibration of sensor and setting the programme NextView 2.5. This provides immediate digitization of information about the

dimensional characteristics of the examined grains. It also caused a risk reduction of human failures within measuring for the reason of incorrect information reading (Fig.2).

Measurements were realized in according with the standard ISO 7256/1 in agricultural conditions where the average speed was 3.09 m.s^{-1} . Measured distances in the row b_{ri} within the evaluation of horizontal seeds placement are classified as following:

- Dual, where distance b_{ri} is
 $0 \odot b_{ri} < 0.5 \text{ (EVR)} - \text{Number } n_0(1)$
- Placement in desiderative distances
 $0.5 \text{ (EVR)} \odot b_{ri} < 1.5 \text{ (EVR)} - \text{Number } n_1(2)$
- Single omission
 $1.5 \text{ (EVR)} \odot b_{ri} < 2.5 \text{ (EVR)} - \text{Number } n_2(3)$
- Double omission
 $2.5 \text{ (EVR)} \odot b_{ri} < 3.5 \text{ (EVR)} - \text{Number } n_3(4)$
- $(k - 1)$ multiple omission
 $(k-1)(\text{EVR}) \odot b_{ri} < (k+0.5) \text{ (EVR)} - \text{Number } n_k(5)$

Where: EVR- is effective distance of plants that is determined by calculation of average value of measured distances b_{ri} . Difference between the truly measured value b_{ri} and calculated effective distance of plants EVR are defined by standard deviation that formulates the variability of plant placement. Above methodology was adopted from mentioned standard and specification according to Páltik et al. (2002).

While study aim is focused on a) characterization of dimensional and shape attributes of seeds, and b) evaluation and comparison of sowing quality of sowing machine with mechanical principle of filling of ladle holes in agricultural conditions, a following seed hybrids was used. Seeds of maize from company Pioneer Hi-Breed Slovakia were used for measurements. It was hybrid PR 37N01. Average harvest of grain was 9.45 t.ha^{-1} during the test. Hybrid has a bountiful harvest in dry and also in wet years. It is designate for the production area of maize. It has a good resistance against the heat stress. Hybrid is able to keep an assimilative area in dry period longer. It grows on a green stalk which is one of the advantages in case of dry period or during the vermin's attack to plants. Quick initial phases of development and good tolerance to cold are reasons for predestination to early sowing. Seeds of big and medium fraction approximately with spherical and angular shape were used within experiments.

Statistical data processing was performed by the analytical software Statistica 10 (TIBCO Software Inc., USA). All of the observed parameters were sampled in min. 20 replication and LSD post-hoc tests were applied to compare mean values. Mean values are presented in the tables (Tab. II, Tab. III and Tab. IV) accompanied with specific standard deviations (in form $\pm \text{s.d.}$). ANOVA test were applied to indicate contrast and statistical differences between tested variables and showed in tables as different letters (a,b,c). Specific parameters are presented in corresponding table legends below tables.

RESULTS AND DISCUSSION

Dimensional and shape attributes of used seeds

Table II shows the average values of length, width, depth and shape coefficients of used seeds are shown in table III. As it was observed that shape attributes have a significant effect on seeding quality and at the end it affects emergency and whole further growth of maize seeds. Shape attributes are defined by the shape coefficients $k_1 - k_4$, if the reciprocal values of coefficients are balanced then seeds will have more ideal spherical shape.

Table II: Dimensional attributes of used seeds

Variant (calibration)		PR 37 N01 LF (big angular)	PR 37N01 MF (medium angular)	PR 37 N01 LR (big spherical)	PR 37N01 MR (medium spherical)
Germinability %		99	98	99	99
Seeds dimensions, n=20	length l	^a 10.98 ±0.12	^a 10.75 ±0.11	^b 10.11 ±0.25	^c 8.89 ±0.14
	width s	^a 8.78 ±0.09	^c 7.87 ±0.12	^a 8.84 ±0.08	^b 8.01 ±0.11
	depth h	^a 4.89 ±0.05	^b 4.69 ±0.08	^c 6.44 ±0.09	^c 6.53 ±0.05

l, s, h - average length, width and depth of seeds; different letters (a,b,c) means significant difference; LSD test; $\alpha=0.05$; s.d. – standard deviation; n=20

Orientation by pushing provides a fast, cheap, and robust way of quickly bringing seeds from a completely random orientation to a known orientation within narrow bounds that could be exploited in a planting mechanism. The results also highlighted that not all hybrids were equally well suited for seed orientation by pushing. Cuboidal or boxy kernel shapes behaved more predictively, rounder shapes exhibited more variability and weaker stable points, a property that could become more important once the unavoidable vibrations of a planter travelling down the field are considered in a seed oriented planting system.

Sowing quality evaluation of selected sowing-machines according to seeds placement

Therefore, precondition of success within this technology is among all others the usage of quality seeds that are placed in row into required depth and at required distance (Testa et al., 2016). The study was observed and concluded at the basis of the analysis of seeds falling trajectories, it was found that the variations of positive differential pressure (Δp) and release angle (θ) had significant effects on seeding uniformity. In order to analyse their effects on horizontal displacement, a coefficient of positive differential pressure variation (X_p) and coefficient of sowing angle variation (X_θ) were proposed and their values calculated (Zhang et al., 2016). In our study it was observed that regularity of the distances changes in dependence on surface roughness, technique of fixing the sowing unit and on working speed (Table IV).

Table III: Shape attributes of used seeds

Variant (calibration) mm ± s.d.		PR 37 N01 LF (big angular)	PR 37N01 MF (medium angular)	PR 37 N01 LR (big spherical)	PR 37N01 MR (medium spherical)
Weight of 1000 seeds, n=15 g ± s.d.		^a 319.79 ±12.16	^b 254.62 ±11.08	^c 359.93 ±13.35	^b 251.64 ±10.93
Shape coefficients, n=20	$k_1 = \frac{l+s}{2} / h$	^a 2.04 ±0.01	^a 2.00 ±0.02	^b 1.49 ±0.2	^c 1.31 ±0.1
	$k_2 = s / h$	^a 1.81 ±0.02	^b 1.69 ±0.01	^c 1.39 ±0.02	^d 1.24 ±0.03
	$k_3 = l / h$	^a 2.26 ±0.03	^a 2.31 ±0.02	^b 1.58 ±0.02	^c 1.38 ±0.03
	$k_4 = l / s$	^a 1.27 ±0.03	^b 1.39 ±0.02	^c 1.16 ±0.02	^c 1.12 ±0.02

k₁, k₂, k₃, k₄- shape coefficients of seeds; different letters (a,b,c) means significant difference; LSD test; $\alpha=0.05$; s.d. – standard deviation; n=20-15, respectively

Standard deviation for sowing maize for distance in a row equal 200 mm should be at sowing system in the range of 18 to 30 mm depending on the operating speed, seeds and type of sowing unit (Páltik et al., 2002). The evaluative criterion of seeds, plants distance variability in the row was value of the standard deviation

of measured distances in the row, a representation of the required and dual seeds placement and percentage representation of the omissions. As it was observed the satisfying quality of sowing maize is considered to be a value of standard deviation from 30 to 40 mm. It was also found out that as a result of these tests, a nominal sowing depth of 60 mm is considered optimum, according

to uniformity of sowing depth and maximum emergence rate index (Özmerzi et al., 2002).

Tests on pneumatic and vacuum precise seeders were carried out to investigate the effects on sowing quality such as up to standard reseeding and miss-seeding etc. The experiment results indicated that the pneumatic seeder is more adaptable to the seed size and shape variation. It was observed that maize seed grading obviously affects the qualified index and reseeding index, but affects miss-seeding index and coefficient of variation little on

condition that all the indexes reach the sowing quality requirements. Moreover, the qualified index of round seeds was 96%, which was higher than a flat seed with the qualified index was 87.4%. There was great distinction between them. The sowing quality of vacuum seeders was not so good whether the seed graded or not, and has not significant influence on the whole indexes of it. The experiment also shows that pneumatic seeders have better performance than the vacuum seeders (Liu et al., 2010).

Table IV: Sowing quality parameters of monitored sowing system within different seeds attributes in agricultural conditions

Machine	Variant (calibration)	Running speed	PVR	EVR	s.d.	Placement of plants				
						PRR	DRR	JV	DV	TV
		m.s ⁻¹ , ±s.d.	mm	mm	mm	%	%	%	%	%
JD Max Emerge XP	JD Max Emerge XP PR 37 N01 LF (big angular)	a3.12 ±0.68	194	195.6	46.95	83.12	3.65	10.12	4.11	1.54
	PR 37 N01 MF (medium angular)	a3.18 ±0.49	194	191.1	50.76	70.9	2.45	15.57	7.37	3.71
	PR 37 N01 LR (big spherical)	a3.30 ±0.52	194	188	47.43	74.33	0.75	17.73	4.9	2.29
	PR 37 N01 MR (medium spherical)	a2.90 ±0.63	194	195.9	48.73	77.15	2.24	14.23	3.74	2.64

Different letters (a,b,c) means significant difference; LSD test; $\alpha=0.05$; PVR - required distance of plants, EVR - effective distance of plants, s.d. - standard deviation, PRR, DRR - required resp. dual placement of plants, JV, DV, TV - single, double, triple omissions

The value of standard deviation is in the rage from 46.95 to 50.76 mm for angular seeds within sowing machine JD MaxEmerge XP. Required seeds resp. plants placement with an average of 83.12 % for angular seeds and 74.33 % for spherical seeds results in better quality of sowing the angular shaped seeds (Table 4). Another evaluative criterion was dual sowing of seeds resp. plants (DRR), where the angular seeds achieved an average of 3.65 % and seeds with approximately spherical shape of 2.24 %. A similar trend was also reached for single omission resp. non-ladle of seeds, sowing with angular seeds reached 3.67 % and spherical seeds 17.73 %.

Özmerzi et al. (2002) emphasises that the importance of operating at the optimum depth has been shown by a number of researchers where sowing too shallow or too deep results in crop performance and yield losses. The speed of drill machinery on seeds distribution within rows was also studied by Ivančan et al. (2004). It was stated that uniformity of seed size and shape has a considerable effect upon the drill precision, notably on the performance of drills with a mechanical drilling unit. The drill operating speed has a decisive influence on the quality and efficiency of the seeding process. Attempts are therefore made to increase maximally the operating speed and still achieve satisfactory sowing quality. The range of optimal speeds should be defined for each crop separately. It was concluded that an increase of operating speed leads to changes of the values of the drill performance indicators. Thus, an increase in operating speed results in a decrease of drilling precision because the intra-row seed spacing becomes larger than that required. The set seed spacing increased from 7.5 to 8.3 cm in parallel with increasing the drilling speed by 1.8 to 5.2 km.h⁻¹,

respectively. The best intra-row distribution of parsley seeds was recorded at the speed of 1.8 km.h⁻¹ (Ivančan et al., 2004).

In addition stated that high-quality seeds can increase the germination rate. Directional seeding can make corn blades grow regularly and enhance ventilation and light energy utilization in the field.

CONCLUSION

Sowing machine JD MaxEmerge XP is able to sow all tested seeds in the required quality; b) Sowing machine JD MaxEmerge XP reached good results for the quality especially when using the angular seeds. It reached the sowing quality expressed by the standard deviation of 46.95 - 50.76 % when seeds PR 37 N01 and for seeds PR 37 N01 it is 47.43 - 48.73 %. According to the results of the field measurements for evaluation of mechanical sowing machine JD MaxEmerge XP in terms of sowing quality with usage of seeds with different dimensional and shape attributes we can express (a,b) following statement: a) Dimensional characteristics of seeds correspond to calibration which is prescribed by the manufacturer. Another evaluative criterion is dual sowing of seeds or plants (DRR), it was reached the average value 3.65 - 2.45 % for angular seeds and the average of 1.54 – 3.71 % for approximately spherical seeds.

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STUDY OF EFFICIENCY OF ADDITIONAL CLEANING OF RAPE SEEDS ON THE ELECTROFRICTIONAL SEPARATOR

ИССЛЕДОВАНИЕ ЭФФЕКТИВНОСТИ ДОПОЛНИТЕЛЬНОЙ ОЧИСТКИ СЕМЯН РАПСА НА ЭЛЕКТРОФРИКЦИОННОМ СЕПАРАТОРЕ

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Abstract: The work is devoted to the increasing the efficiency of preparation for sowing of rapeseed material by the additional cleaning. The obtained results of researches show that due to additional processing of seeds it is possible to improve weight and dimensional characteristics of seeds of rape sowing material by 20%. It was also found that the mass of individual seeds is in a clear correlation with the seed size.

KEYWORDS: RAPE SEEDS, ELECTROFRICTION SEPARATOR, ADDITIONAL CLEANING, MASS AND DIAMETER OF SEED

1. Введение

В семеноводстве масличных культур, в частности озимого рапса, важную роль играет вопрос улучшения качества посевного материала, который не всегда соответствует требованиям существующих стандартов из-за засоренности сложноотделяемыми примесями и семенами, которые имеют низкий биологический потенциал [1 - 10]. Использование такого посевного материала не позволяет реализовать в урожае потенциальной продуктивности сорта культурного растения.

Большинство семяочистительных машин не в состоянии провести процесс сепарирования, который бы позволил получить посевной материал без разного рода примесей и нежизнеспособных семян. В связи с этим актуальным является поиск способов и технических средств, которые бы существенно повышали эффективность послеуборочной обработки данной культуры.

2. Предпосылки и средства для решения проблемы

Исследованиям процесса послеуборочной обработки семян на электрофрикционном сепараторе посвящено ряд научных трудов [3, 7, 9, 10]. Определения оптимальных значений параметров процесса сепарирования семян озимого рапса на электрофрикционном сепараторе осуществляли путем определения содержания некачественных семян в получаемом в процессе сепарирования посевном материале.

На основе анализа полученных результатов было установлено, что достичь минимального содержания некачественных семян ($K_{min} = 4\%$) возможно при условии угла наклона сепарирующей плоскости $\alpha = 9$ град., скорости ее движения $V_n = 0,07$ м/с и напряженности электрического поля в рабочей зоне сепаратора $E = 1,8 \dots 2$ кВ/см [3, 7, 9].

Благодаря электросепарированию семян озимого рапса на электрофрикционном сепараторе удастся уменьшить количество поврежденных семян в посевном материале и довести их содержание до 2 ... 4%, что на 14 ... 16% меньше по сравнению с содержанием некондиционных семян в посевном материале озимого рапса [3, 7].

Однако требованиями стандартов к семенам масличных культур, и рапса в частности, установлены ограничения по максимальному содержанию засорителей (чистоты семян) и массы тысячи семян (m_{1000}). К сожалению, в работах [2, 3, 4, 6, 7, 8, 9] эти данные не приведены. Поэтому целью наших исследований было установление корреляционной связи между

массой и геометрическими параметрами (размерами) семян в результате их сепарирования на электрофрикционном сепараторе.

3. Решение рассматриваемой проблемы

Целью исследований было подтвердить эффективность сепарирования семян на электрофрикционном сепараторе с точки зрения отбора семян с максимальными весовыми и размерными характеристиками, а также установить корреляционную связь между массой и размерами отдельных семян.

Для этого необходимо было провести сепарирования семян озимого рапса и получить образцы семян для исследования их размерных и весовых характеристик. Экспериментальные исследования проводили на электрофрикционном сепараторе представленном на рис. 1. Он состоит из рамы 1, бункера-питателя 2, сепарирующего узла 3, механизма регулировки наклона сепарирующего узла 4 и электрооборудования 5.



Рис.1. Общий вид электрофрикционного сепаратора

В результате сепарирования нами было получено два продукта сепарации, а именно посевной материал и отходы. Из них отбирались пробы для определения размеров и массы отдельных семян.

Размеры (диаметр) семян рапса определяли с помощью электронного микроскопа (рис. 2).



Рис. 2. Электронный микроскоп

Измерение массы отдельных семян рапса проводили на лабораторных весах МYA 11.4Y "Radwag" (Польша) с дискретностью 0,000001 г (рис. 3).



Рис. 3. Весы лабораторные МYA 11.4Y "Radwag"

Исследования проводились для семян озимого рапса сорта Дангал первой репродукции.

4. Результаты и дискуссия

В результате статистической обработки экспериментальных данных установлены минимальные, максимальные и средние арифметические значения массы и диаметров отдельных семян рапса (см. табл.)

Таблица 1

Результаты исследований

№	Образец	Масса семян, мг	Среднее	Диаметр семян, мм	Среднее
1	Контроль	3,6...8,1	5,42	1,75...2,9	2,25
2	Посевной материал	4,9...8,3	6,76	2,25...3,15	2,85
3	Отходы	3,3...5,2	4,13	1,65...2,1	1,75

Анализируя данные табл. 1 видно, что по сравнению с контрольной пробой семян средняя масса и размеры отдельных семян в посевном материале выросла на 19,8% и 20% соответственно, что подтверждает эффективность работы сепаратора.

Для определения корреляционной связи между массой и размерами семян рапса была произведена статистическая обработка результатов с помощью программы STATISTICA 10, в результате которой были определены коэффициенты корреляции и построены зависимости массы семян от их размеров (рис. 4 - 6).

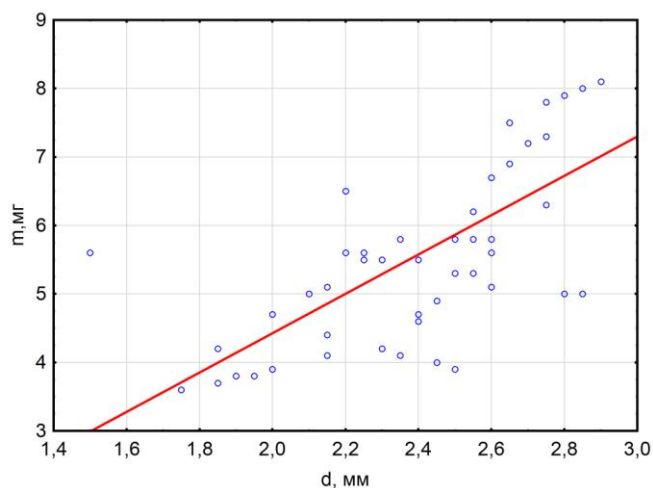


Рис. 4. Зависимость массы семян в контрольной пробе от их размеров

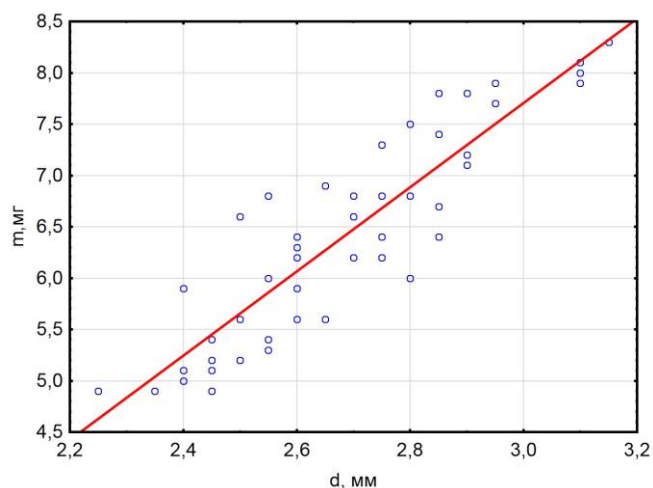


Рис. 5. Зависимость массы семян в посевном материале от их размеров

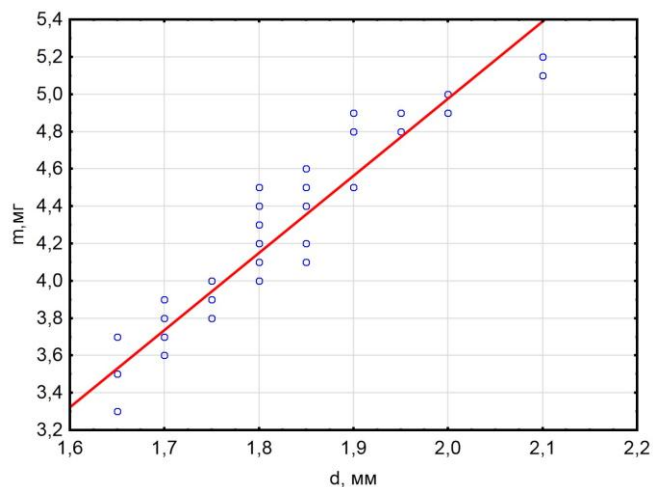


Рис. 6. Зависимость массы семян в отходах от их размеров

В результате определения коэффициента корреляции установлено, что для семян контрольной пробы рапса он был равен 0,71. Для продуктов сепарирования исследуемый коэффициент возрос до 0,9 у посевного материала и 0,95 у отходов сепарации.

Значение коэффициента корреляции выше 0,6 характеризует чёткую корреляционную связь между размерами семян и его массой. Увеличение данного показателя в результате дополнительной обработки семян рапса на электрофрикционном сепараторе указывает на эффективность его работы и возможность разделения однокомпонентной семенной смеси рапса по массе и размерам отдельных семян.

5. Заключение

Полученные результаты исследований свидетельствуют об эффективности дополнительной обработки семян на электрофрикционном сепараторе, поскольку благодаря ей удалось повысить среднюю массу отдельных семян в посевном материале. Большие значения коэффициентов корреляции для посевного материала и отходов сепарации по сравнению с аналогичным показателем для контрольной пробы (необработанных семян) свидетельствуют о четкой зависимости между размерными и весовыми характеристиками семян.

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THE THEORETICAL ESTIMATION OF THE TRACTOR DRIVE WHEELS SLIPPAGE WITH THE VARIABLE TIRE INFLATION PRESSURE.

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Abstract: The main tractor parameters that defy suitability of tractors to perform traction work is traction power and driven axle wheel-slide. Different mass tractors wheel-slide on the same soil differs and depends on the vertical load (G), which falls on the driven axle wheels. When choosing tractors and agriculture implements it is needed to foresee the formed implements traction and economical parameters. To achieve this goal mathematical modeling methods are used. The purpose of mathematical modeling is to determine traction characteristics of the tractor, taking into account various factors. At this moment there are tractor traction models created, that describe various agriculture and field tasks. Most Mathematical models mainly assess the hardness of the soil and driven wheel geometrical parameters. However, one of the most important parameters in assessing the suitability of a tractor unit is the tractors driven wheels slide. To decrease wheel-slide and to increase the driving wheels grip with soil there are different methods being used: Ballast weights, use of twin wheels and changeable air pressure in the tires. However current mathematical models do not consider the change of air pressure in tires. A mathematical module was created, which takes into calculations the driven wheels wheel-slide having different air pressure in the tires.

Keywords: TRACTOR IMPLEMENT, MATHEMATICAL MODELING, TRACTION POWER, WHEEL-SLIDE, AIR PRESSURE

1. Introduction

Tractors wheels slippage is the most important tractors and it's implements forming energetic – economic and ecologic parameter. (Osinenko et al., 2015) It is one of the main parameters that shows the tires and the roads (soils) surface interaction. (Lee et al. 2016; Molari et al. 2012; Grečenko, and Prikner 2014). Different mass tractors wheel slippage on the same soil differs and depends on the vertical load (G), which falls on the driven axle wheels (Damanauskas and Janulevičius 2015, Zoz 1972) In agrotechnical requirements allowed slippage limits are set. When ploughing stubble ground wheel slippage is recommended to be around 8...12%. Wheeled tractor permissible wheel slippage – up to 15 %. When there is wheel slippage, a higher traction power can be achieved. Although when it is too high or too low energy loss rises. (Batitatio and Diserens 2013, Lacour et al. 2014). A tractor can achieve the highest traction power when there is 15...20% wheels slippage. Although due to the negative effect to the soil, the maximum allowed wheel slippage for wheeled tractors must not exceed 15%. (Lee et al. 2016; Molari et al. 2012).

At this moment, one of the most popular ways to decrease wheel slippage is changeable air pressure in the driven tractor tires. It is advisable to decrease air pressure in the driven tires, then the tire deforms, the area of its support increases, and the grip with surface improves. In addition, high wheel slippage reduced tractors coefficient of efficiency (performance) and cost-effectiveness.

The aim of this work was to find a mathematical formula for theoretical calculation (prognosis) of tractors driven wheel slippage, while evaluating air pressure in the tires.

2. Results and Discussion

When compiling vehicles and aggregates and preparing them for work, advance calculations (forecasting) of wheel slippage is an important task to determine energetic and economic parameters.

The tractor pulling forces expositive a non-dimension form: the net traction coefficient of the tractor (c_T), the gross traction or torque coefficient (c_G), and the total motion resistance coefficient of the tractor (c_W) offer to evaluate tractors wheel slippage using the formula:

$$s_t = s_{lim} \left[1 - \left(1 - \frac{F_T}{F_T^{lim}} \right)^b \right] \quad (1)$$

where: s_t – theoretical wheel slippage, F_T^{lim} – maximum traction power at the limit wheel slippage, kN ; F_T – tractors traction power, kN; s_{lim} – limit wheel slippage (wheeled tractors, $\delta_{rib}=40\%$). b –grade indicator.

Since in almost all cases the mathematical model's error is close to 20% and, in some cases, exceeds, also this mathematical model does not evaluate the air pressure in the tires.

From the analysis of the tractors experimental studies wheels slippage dependency from traction power on various tires air pressures, it was decided, that instead of using constant grade indicator (b) to use the indicator as a changing value, which reflects the air pressure in the tractors tires.

We can call this size air pressure in the tires coefficient (k_p). Then we can rewrite the formula (ar naudoti expression) as follows:

$$s_t = s_{lim} \left[1 - \left(1 - \frac{F_T}{F_T^{lim}} \right)^{k_p} \right] \quad (2)$$

The main goal of this mathematical module is to calculate tractors wheel slippage, according to maximum nominal traction power F_T^{lim} . Nominal traction power is determined experimentally when wheel slippage is at 40%.

Experimental tests were performed to determine the coefficient (k_p) depending on the air pressure in the tires. Tractors driven wheel slippage was determined by experimental tests according to standard methodology and calculated according to the formula (Gray et al.,2015; Janulevičius et al.,2017):

$$S = \frac{v_t - v}{v_t} \quad (3)$$

here v_t – theoretical speed; v – actual speed.

Experiments were done using tractors "CASE IH Farmall U Pro 115", "New Holland T5070". Tractor "CASE IH Farmall U Pro 115" and "New Holland T5070" main technical characteristics are given in table nr.1. During the tests, the air pressure was changed at the front and rear tires of the "CASE IH Farmall U Pro 115" and "New Holland T5070". The air pressure in front and rear tires was the same. The tests were done with maintaining the air pressure at 80, 120, 140, 160, 200, 240 kPa. The tests have been conducted with the front axle on and off and the rear differential lock permanently disabled. The load on the tested tractors was generated by pulling tractors "Zetor 10540" and "MTZ 82". Two tractors were used to fully load the tested tractor "CASE IH Farmall U Pro 115" or "New Holland T5070", which were connected with a rigid towing bar, which was horizontal to the surface of the road. The test execution scheme is shown in figure 1.

Table 1: "CASE IH Farmall U Pro 115" and "New Holland T5070" technical data:

Technical data	CASE IH Farmall U Pro 115	New Holland T5070
Engine power, kW	84	83
Weight of the tractor, kg	4336	4350
Wheelbase, mm	2380	2350
Front tires	Trelleborg 480/65 R24	Michelin Multibib 40/65R24
Rear tire	Trelleborg 540/65 R38	Michelin Multibib 540/65R24
Weight of the front axle, kg	2004	1990
Weight of the rear axle, kg	2896	2260

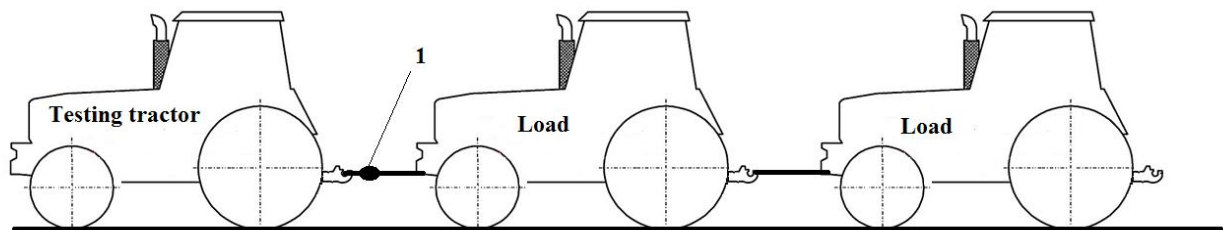


Fig 1. Test execution scheme. 1- force measuring.

Coefficient (k_p) dependence on the air pressure of the tractor "Case Farmall 115 U" (tires "Michelin Multibib 540/65R34") and tractor "New Holland T5070" (tires "Trelleborg 540/65 R38") tires using 4X4 driven wheels is given in figure 2.

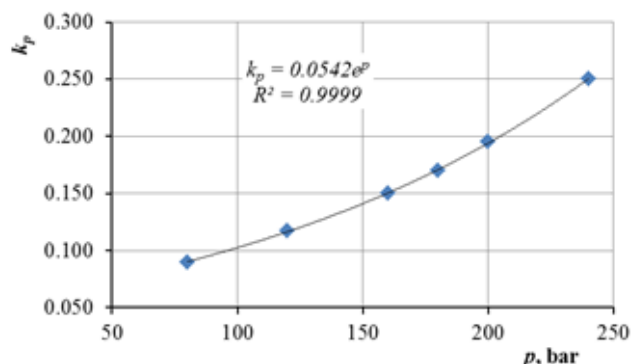


Fig 2. Coefficient (k_p) dependency from air pressure in the tractors tires.

Coefficient (k_p) was determined using tractors with 4X2 driven wheels. The difference between the two tractors results didn't reach 5%. In figures 2 and 3 tractors "New Holland T5070" test results are displayed.

As we can see from figure 2 air pressure coefficient (k_p) directly depends from air pressure in tractor tires according to exponential dependence:

$$k_p = 0,0542e^p$$

(4)

here: p – air pressure in tractors driven wheel tires kPa using tractors 4X2 driven wheels.

With a pressure of 80 kPa in tractor tires, the coefficient (k_p) for air pressure in tires is 0,090. When the nominal air pressure in the tractor tires is 180 kPa, the coefficient of air pressure in the tires is considered to be nominal and equals 0,170. (Fig. 2).

After assigning corresponding values of coefficient (k_p) to the formula (3) and after calculations, we obtain wheel slippage theoretical dependency from traction power. A comparison of these dependencies with experimental curves is presented in Figure (3).

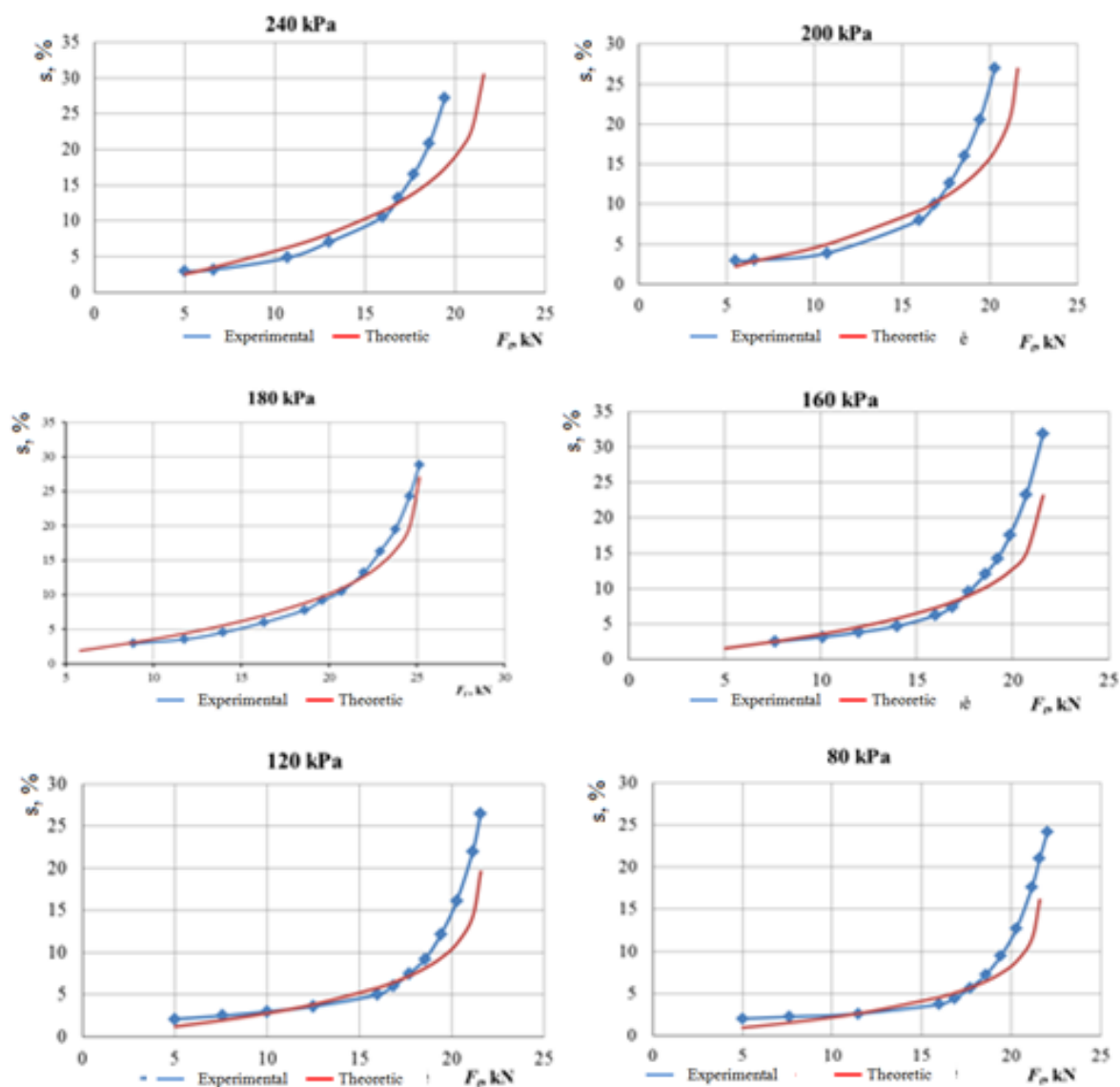


Fig 3 Tractors “New Holland T5070” wheel slippage and traction power theoretical and experimental dependencies using tractors 4X2 driven wheels

Conclusions

1. Having done literature analysis, it has been found that, known tractors wheels slippage determining mathematical modules, cannot directly take into consideration the air pressure in tires.
2. To calculate traction and wheel slippage, of the mathematical equation, coefficients (k_p) dependency from air pressure in tractors tires was found(set).
3. A mathematical model was made for the tractors “New Holland T5070” wheel slippage prediction from traction power, which is appropriate having small or medium loads, up to 17-18 kN traction power.

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ВЕНТИЛЬНЫЙ ЭЛЕКТРОГЕНЕРАТОР С ПРИВОДОМ ОТ ВАЛА С РАЗЛИЧНОЙ ЧАСТОТОЙ ВРАЩЕНИЯ

VENTILATED ELECTRIC GENERATOR WITH DRIVE FROM SHAFT WITH VARIOUS ROTATION FREQUENCY

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Резюме: В статье рассматривается вопрос поддержания напряжения на выводах синхронного генератора в допустимых пределах при различной частоте вращения его ротора. Предложены способы последовательного и параллельного соединения ветвей обмоток якоря, соответственно, при снижении или повышении частоты вращения ротора относительно номинального значения. Приведены электрические схемы и общий вид испытательного стенда. Получены экспериментальные данные и построены кривые, характеризующие изменения выходного напряжения при различном соединении ветвей обмоток якоря.

КЛЮЧЕВЫЕ СЛОВА: ЭЛЕКТРОГЕНЕРАТОР, ЧАСТОТА ВРАЩЕНИЯ, ОБМОТКА СТАТОРА, ВЕНТИЛЬ, НАПРЯЖЕНИЕ, ЧАСТОТА ТОКА.

1. Введение

Развитие рыночного хозяйства предъявляет повышенное требование к бесперебойному электроснабжению сельскохозяйственного производства. Перерывы в электроснабжении наносят сельскому хозяйству существенный материальный ущерб и дезорганизуют ход технологических процессов.

Для повышения надежности электроснабжения желательно использовать сетевое резервирование. Однако оно не всегда обеспечивает бесперебойное электроснабжение, в частности, при неблагоприятных погодных условиях (гололед, ураганные ветры и т.п.). Кроме того, большинство сельских потребителей Казахстана удалены на значительные расстояния и обеспечиваются электроэнергией по радиальным линиям электропередачи значительной протяженности, что затрудняет применение сетевого резервирования. В этом случае эффективным является применение традиционных и нетрадиционных автономных источников электроэнергии.

Неоспоримым является применение нетрадиционных источников энергии, среди которых для Казахстана особый интерес представляют: ветровые, солнечные и малые гидроэлектростанции. Вместе с тем, эти источники энергии пока не получили должного распространения из-за сложности технических и экономических решений.

В настоящее время, при перерывах в электроснабжении, в качестве основного автономного резервного источника электроэнергии, применяют дизельные электростанции и электроагрегаты (ДЭС), передвижные электростанции (ПЭС) и электрогенераторные установки с приводом от вала отбора мощности трактора (ЭГУТ).

2.1. Предпосылки и решения проблемы

В настоящее время, в Казахстане, для автономного и резервного электроснабжения отдельных объектов преимущественно используют дизельные электростанции и электроагрегаты (ДЭС), рыночные цены которых не устраивают многочисленные малые и средние предприятия.

На некоторых сельскохозяйственных объектах нашли применение электрогенераторные установки с приводом от трактора (ЭГУТ). Ограниченность применения ЭГУТ заключается в том, что они предназначены для работы от вала отбора мощности (ВОМ) с конкретной частотой вращения (540 или 1000 об/мин) и имеют различные механизмы преобразования (зубчатые редукторы, массивные шкивы и т.п.) частоты вращения вала электрогенератора. Номинальная

частота распространенных синхронных генераторов (СГ) составляет 1500 и 3000 об/мин [1]. Следует отметить, что механизмы преобразования частоты вращения увеличивают вес ЭГУТ, снижают надежность конструкции и требуют дополнительного технического обслуживания. Наряду с этим, с переменной частотой вращения ВОМ трактора возникают проблемы поддержания необходимого напряжения и частоты тока на выводах электрогенератора. Решение этих задач за счет изменения диаметров ведущего и ведомого шкивов ЭГУТ и зубчатого практически невозможно.

Следует отметить, что в большинстве хозяйств могут иметь место источники механической энергии с различной частотой вращения ВОМ (трактора с номинальной частотой вращения 1000 и 540 об/мин, гидроустановки с частотой вращения 50 - 150 об/мин, ветроустановки с различной частотой вращения 200-700 об/мин. и т.д.) [1,2 и др.].

Поддержание стабильного напряжения и частоты тока в указанных условиях определяет необходимость исследования и формирования рациональных рабочих режимов ЭГУТ, с последующей формулировкой требований как к системе возбуждения, так и к входным параметрам преобразователя частоты СГ.

При переменной или разной частоте вращения возникает вопрос обеспечения постоянства частоты переменного тока. Можно выделить два основных решения: применение синхронизированных асинхронных генераторов и применение вентильных асинхронных и синхронных генераторов. По мнению авторов, одним из перспективных решений является вариант с использованием вентильного синхронного генератора и автономным инвертором, который преобразует постоянный ток от выпрямителя в переменный ток с постоянной частотой. Такие установки последнее время называют инверторными генераторами [3].

Как правило, традиционные электрогенераторы без дополнительных устройств не могут обеспечить поддержание величины напряжения. При снижении частоты вращения ниже 53-55 % обеспечить поддержание напряжения из-за насыщения магнитной цепи генератора проблематично [3]. Чем шире диапазон изменения частоты вращения, тем ниже КПД генератора. Использование инверторов с поддержанием постоянства величины напряжения при изменении входного напряжения может расширить рабочий диапазон. Однако, вопрос поддержания КПД генератора такое решение не обеспечивает.

Практический интерес представляет разработка универсального генератора, вращаемого от ВОМ с различной и переменной частотой вращения и при этом поддерживать требуемую величину напряжения и удовлетворительного КПД без использования специализированных устройств.

В данной статье предлагаются способы создания вентильного электрогенератора (ВЭГ) с приводом от различных источников механической энергии (ветро- и гидроустановок, тракторов и т.д.) независимо от частоты вращения их ВОМ. При этом отпадает необходимость применения каких-либо редукторов для преобразования частоты вращения.

Согласно теории электрических машин, выходное напряжение генератора можно определить по выражению [4.5]:

$$(1) \quad U_r = 4,44 \cdot f \cdot W_{\phi 1} \cdot \Phi = 4,44 \frac{p \cdot n}{60} W_{\phi 1} \cdot \alpha \cdot \tau \cdot l \cdot B_{\delta} = \\ = 4,44 \frac{p \cdot \tau \cdot l \cdot \alpha}{60} W_{\phi 1} \cdot n \cdot B_{\delta} = K_r \cdot W_{\phi 1} \cdot n \cdot B_{\delta}$$

где $W_{\phi 1}$ - число эффективных витков в одной параллельной фазной ветви;

B_{δ} - индукция в воздушном зазоре;

K_r - конструктивная постоянная генератора;

p - число пар полюсов;

α - коэффициент полюсного перекрытия;

τ - полюсное деление;

l - длина сердечника;

f - частота тока;

n - расчетная частота вращения.

Из выражения видно, что при снижении частоты вращения в два раза потребует соответствующего увеличения индукции (тока возбуждения), что в свою очередь приведет к росту потерь в обмотке возбуждения и насыщению магнитопровода. Согласно рекомендации по проектированию электрических машин индукция в воздушном зазоре принимается в диапазоне 0,6-0,7 Тл, при этом индукция в зубцах обычно достигает значения 1,3-1,6 Тл, а насыщения электротехнической стали в диапазоне 1,8-1,9 Тл. [4.5]. Следовательно, отсутствует возможность увеличения индукции в воздушном зазоре пропорционально изменению частоты вращения. Проектирования машины с пониженным значением индукции приведет к завышенным массогабаритным показателям и к росту влияния реакции якоря на нагрузочные характеристики. Это следует из выражения, определяющего коэффициента Арнольда [4.5]:

$$(2) \quad \frac{D^2 l_1 n}{P'} = \frac{6,1 \cdot 10^7}{(\alpha_i k_{\phi} k_o AB_{\delta})} = C_A$$

где C_A - машинная постоянная Арнольда;

P' - расчетная мощность;

k_{ϕ} - коэффициент формы кривой поля;

k_o - обмоточный коэффициент;

A - линейная нагрузка.

Как следует из выражения (1) при заниженной индукции нужно увеличить количество витков в обмотке, а это в свою очередь повлияет на величину реакции якоря, как следует из выражения [4.5]:

$$(3) \quad F_p = 0,9 \cdot m \frac{W_1 \cdot k_o}{p} \cdot I_{\phi}$$

где m - число фаз;

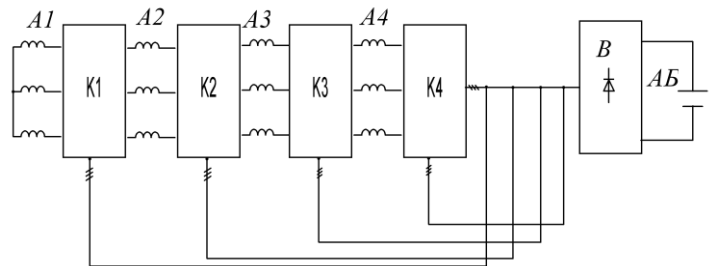
I_{ϕ} - фазный ток.

Из анализа следует, что все общеизвестные и традиционные способы расширения диапазона рабочих частот вращения ухудшают массогабаритные показатели машины.

Группой ученых Алматинского университета энергетики связи разработан электромеханический преобразователь с широким диапазоном изменения частоты вращения и

мощности, способный поддерживать постоянство величины напряжения при кубической зависимости мощности от частоты вращения. При этом сохраняет удовлетворительное значение КПД в диапазоне от 0,75 до 0,94 [6]. Идея разработки заключается в изготовлении якорной обмотки генератора из нескольких параллельных ветвей подключенных к своему выпрямителю и образующих вентильные блоки. Вентильные блоки объединены в коммутируемый выпрямитель с возможностью переключения последовательно, параллельно и последовательно параллельно. Фактически в одном корпусе совмещается несколько генераторов на различные рабочие режимы. Коммутируемый выпрямитель является дискретным регулятором, а плавное поддержание напряжения производится током возбуждения. Коммутируемый выпрямитель производит динамическое переключение ступеней под нагрузкой. Реализация динамического переключения без коммутируемого выпрямителя будет затруднительно. Поэтому при работе в составе ветряной электростанции большое количество параллельных ветвей с минимальным количеством коммутирующих контакторов схемой оправдывается.

Предлагаемый ВЭГ приведен на фиг. 1. ВЭГ имеет якорную обмотку, состоящей из «п» параллельных ветвей (в частном случае $n=2$) и «м» коммутирующих аппаратов. Начало первой параллельной ветви соединены вместе и образует соединение «звезда». Концы и начало остальных параллельных ветвей через коммутирующие аппараты соединены между собой и с выпрямителем через коммутирующие аппараты. Концы одной из параллельных ветвей подключены к выпрямителю.



Фигура 1. Вентильный электрогенератор

Изменение схемы соединения параллельных ветвей производится на стороне переменного тока полупроводниковыми коммутаторами.

При высокой частоте вращения коммутирующие аппараты соединяют все параллельные ветви по схеме «звезда» и параллельно между собой, каждая из которых подключена к выпрямителю. При работе на пониженной частоте вращения коммутирующие аппараты соединяют параллельные ветви последовательно между собой. Количество переключений зависит от количества групп параллельных ветвей.

Величина напряжения ВЭГ с двумя параллельными ветвями допускает две ступени переключения схемы соединения ветвей обмотки и определяются по выражениям:

- первая ступень, на самой высокой частоте вращения все ветви параллельны [4.5]:

$$(4) \quad U_r = U_1 + U_2 = K_r \cdot W_{\phi 1} \cdot n \cdot B_{\delta}$$

- вторая ступень ветви соединены последовательно;

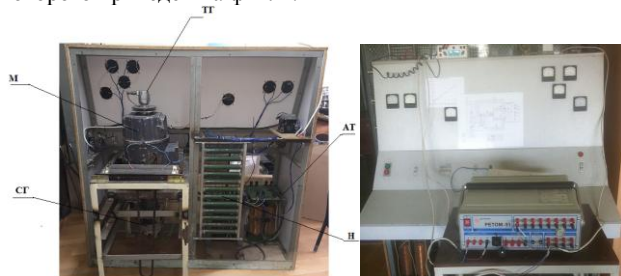
$$(5) \quad U_r = U_1 + U_2 = 2W_{\phi 1} (K_r \cdot n \cdot B_{\delta})$$

Выражение (1) можно переписать в следующем виде:

$$(6) \quad U_r = U_1 + U_2 = a \cdot W_{\phi 1} \cdot K_r \cdot n \cdot B_{\delta}$$

Переключение схемы соединения параллельных ветвей равносильно изменению числа витков в параллельных ветвях якорной обмотки. Согласно выражению (1) при увеличении

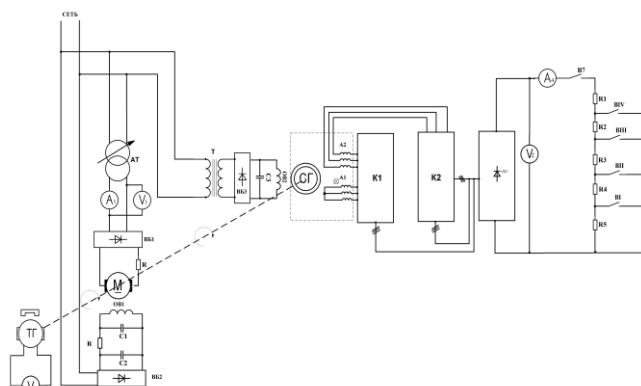
витков в обмотке понижается величина индукции, требуемая для поддержания напряжения. Тестирование генератора приводилось на экспериментальном стенде лаборатории Алматинского университета энергетики и связи, общий вид которого приведен на фиг. 2.



Фигура 2. Внешний вид испытательного стенда

В качестве генератора используется автомобильный генератор типа Г-286В (СГ). Основные технические характеристики генератора типа Г-286В, номинальное напряжение 14 В, номинальный ток 50 А, частота вращения с регулятором напряжения при самовозбуждении и напряжения 13В без нагрузки 650 об/мин, при токе нагрузки 20А при 750 об/мин. Максимальный ток нагрузки 75А а при 5000 об/мин. Генератор вращается приводным двигателем (М) постоянной тока П32М, мощностью 2,2 кВт, с номинальной частотой вращения 1500 об/мин. Якорная обмотка генератора перемотана и выполнена с двумя параллельными ветвями на напряжение 24 В. Обмотка возбуждения не изменялась.

Принципиальная схема испытательного стенда приведена на фиг. 3.

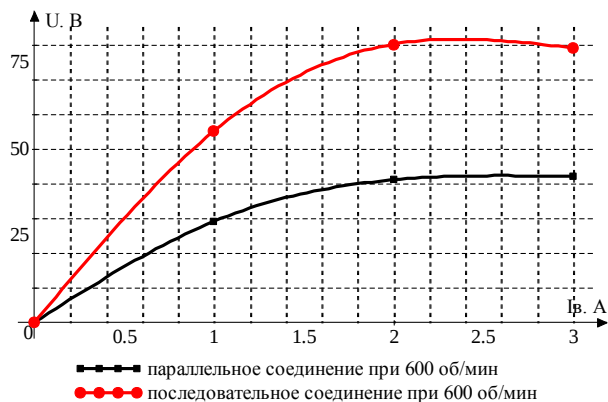


Фигура 3. Принципиальная схема испытательного стенда

Согласно фиг. 3 вал отбора мощности моделировался двигателем постоянного тока независимого возбуждения П-32М (М), мощностью 2,2 кВт и частотой вращения 1500 об/мин. Регулирование частоты вращения генератора производилось регулированием напряжения двигателя через автотрансформатор. Согласно принципиальной схеме синхронный генератор СГ соединен с валом приводным двигателем М который. Угловая скорость измеряется с тахогенератором ТГ. Обмотки СГ разделен на две параллельные ветви А1 и А2, которые через коммутирующие ключи К1 и К2 подключены к выпрямителю.

Напряжение в параллельных ветвях замерялось на стороне постоянного напряжения на выводах выпрямителей и на стороне переменного напряжения на выводах параллельных ветвей.

На основе экспериментальных данных на фиг. 4 приведены специальная характеристика холостого хода.



Фигура 4. Специальная характеристика холостого хода

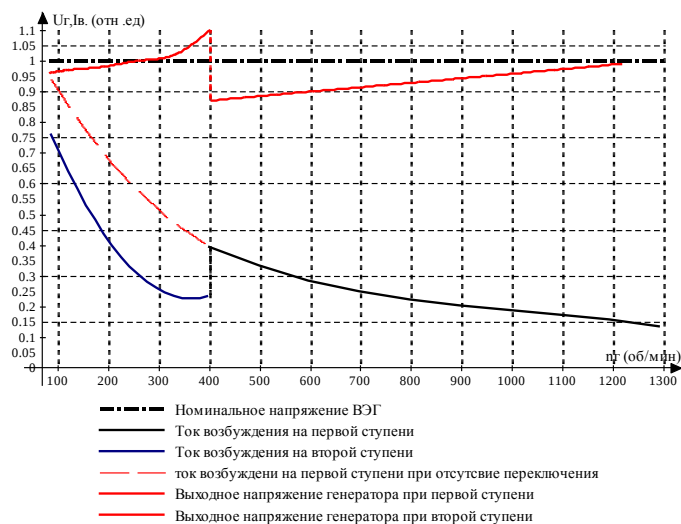
Из фиг. 4 видно, что переключение ветвей дает возможность увеличить или уменьшить величину выходного напряжения ВЭГ в соответствии с потребностями. Так, переключение двух ветвей с параллельного соединения на последовательное увеличивает напряжение в два раза. И наоборот. При этом частот вращения остается постоянной.

Характеристика холостого хода ВГ определяется зависимостью тока возбуждения I_e от частоты вращения n_r , при поддержании требуемого выходного напряжения $U = const$ и $I_n = 0$.

На фиг. 5 приведена специализированная характеристика холостого хода, которая построена при постоянном напряжении (линия 1) и изменении частоты вращения от 1250 до 100 об/мин.

На рисунке 5 приведены: линия напряжения (линия 1) и тока возбуждения на двух ступенях (кривые 2,3). Пунктирная линия отображает изменение тока возбуждения при отсутствии переключения. Масштабы напряжения и тока возбуждения генератора приведены в относительных величинах. Напряжение на протяжении всего диапазона изменения частоты вращения поддерживалось около номинального значения. Ток возбуждения при частоте вращения в 100 об/мин достигает 3 А, то есть свое максимальное значение.

В эксперименте использовался автомобильный генератор Г286В, который был перемотан на расчетную скорость 1200 об/мин. Количество параллельных ветвей принято $a=2$, мощность 300 Вт. При количестве ветвей « $a=2$ » количество ступеней равно 2.



Фигура 5. Специальная характеристика холостого хода перемотанного автомобильного генератора

Из фиг.5. видно, что при снижении частоты вращения ВЭГ снижается выходное напряжение. Для поддержания выходного напряжения на требуемом уровне необходимо увеличивать ток возбуждения. Однако в последнем случае возникает опасность перегрева обмоток тока возбуждения. Поэтому при определенной частоте вращения (примерно 400 об/мин) производилось переключение ветвей обмоток статора на последовательное соединение. При этом напряжение ВЭГ увеличилось примерно на 10-12%, а ток возбуждения уменьшился в такой же пропорции.

По результатам эксперимента видно, что напряжение генератора поддерживается постоянным на всем диапазоне изменения частоты вращения.

Пересчет генератора производился на расчетную частоту вращения 1200 об/мин. и при пониженной расчетной индукции в воздушном зазоре на 0,2 Тл. Диапазон изменения тока возбуждения от 0,4 до 3 А. На графике за единицу принято максимальное значение тока возбуждения 3А. Кривая тока возбуждения на первой ступени при отсутствии переключения в относительных единицах изменяется в диапазоне от 0,15 до 0,9. Широкий диапазон объясняется заниженным значением расчетной индукции. Переключение на следующую ступень снижает, согласно концепции, значение тока возбуждения в 2 раза (пунктирная кривая линия, соединяющая кривую 2 и 3).

Таким образом, схемное решение ВЭГ, вращаемого от ВОМ с различной или переменной частотой вращения позволяет поддерживать требуемую величину напряжения и удовлетворительного КПД.

3. Заключение

В статье предложено схемное решение разработки ВЭГ, способного выдавать напряжение и частоту тока в заданных пределах при вращении от ротора с различной частоты вращения. Установлено, что при снижении частоты вращения ВОМ ветви обмоток статора целесообразно соединять последовательно, а при увеличении частоты вращения – параллельно. Тестирование подтвердило правильность теоретических выводов и работоспособность данного электрогенератора.

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RESEARCH OF USING THE ALTERNATIVE FUELS FOR IMPROVING THE INDICATORS OF THE TRACTOR DIESEL WORK

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Topicality: The urgency of the issue under consideration is stipulated by large-scale international research and is confirmed by the Russian Federation Government Decree No. 1-P of 08.01.2009 "Main Directions of State Policy in the Sphere of Energy Efficiency Improvement on the Basis of Using Renewable Energy Sources for the Period to 2020" and the Decree of the President of the Russian Federation from July 7, 2011 № 899 on priority directions of development of science, technology and technics in the Russian Federation.

According to the analytical agency "AUTOSTAT", as of 2017 there are 3.7. million of trucks in the Russian Federation, of which the ecological standards "Euro-4" (and above) correspond only 14%, and annually are used more than 20 million tons for freight transport, which in the amount of processing oil to diesel gas oil is more than 70 million tons of diesel gas oil. From the above, it can be concluded that in order to solve the fuel issue and preserve the environment, it is necessary to study the performance of the diesel engine using diesel composite fuel and transfer of transport to alternative fuels.

As a raw material for the production of alternative fuels is attractive biomass, such as oilseeds and waste from processing plants.

As a result of the growing demand for ricinic oil by industry, it is necessary to study its application as a component for the production of diesel composite fuel.

However, the viscosity of ricinic oil is higher than that of diesel gas oil, so it can only be used as an additive to low-viscosity components. As a result of the addition of diesel fuel and bioethanol to ricinic oil, with preliminary preparation, decreases viscosity and are improved low-temperature properties. This mixture can be suitable for use on transport-technological machines, the point of view of qualitative spraying in areas with a moderate climate.

Based on the thermal calculation of the engine by the methods of Prokopenko R.M., Khorosh A.I., Bashirova R.M., are determined the parameters of the working cycle of the engine D-240 and are presented below the dependences on the concentrations of the components.

The article compares the obtained results with the indicators of traditional diesel gas oil.

KEY WORDS: DIESEL COMPOSITE FUEL (DCF), DIESEL, INTERNAL COMBUSTION ENGINE, MIXTURE, POWER, ECONOMY, ALTERNATIVE TYPE OF MOTOR FUEL, RICINIC OIL, CASTOR OIL, BIOETHANOL.

1. INTRODUCTION.

In modern times, the development of motor – and - tractor technics is greatly influenced by:

- toughening of environmental norms on emissions of toxic substances;
- methods for increasing energy efficiency.

Today, the Russian Federation produces more than 100 million tons of motor fuel produced in the traditional way from oil (Figure No. 1) [2];

At the same time, the dynamics of reserves growth due to revaluation and geological exploration remains at the level of 2007 and 2008, which means that with an intensive increase in the number of stationary and mobile consumers of petroleum products, the demand for petroleum products may exceed the growth dynamics of reserves. As a result, there will be an intensive decrease in the world reserves of oil deposits. [2]

Therefore, one of the main ways of developing agricultural and logging production can be considered a partial transfer to alternative fuels from biomass. This will solve the problem of replacing petroleum fuel, significantly expand the resource base for fuel production, facilitate the solution of fuel supply issues for vehicles and fixed installations remote from large settlements. This can reduce the prime cost of production, subject to the independent production of components of the DCF or in cooperation with other manufacturers. [13,14]

As a raw material is attractive biomass for obtaining an alternative fuel, such as oilseeds and waste from the processing industry.

As a result of growing demand for ricinic oil, which is used in many industries, and the resumption of research in the Russian Federation aimed at selecting and processing castor oil plant, there is a need to investigate ricinic oil and bioethanol as

biocomponents for the production of diesel composite fuel. [14,15]

However, the viscosity of ricinic oil is higher than that of diesel gas oil, therefore ricinic oil can only be used as an additive to low-viscosity components. As a result of the addition of diesel fuel and bioethanol to ricinic oil, with preliminary preparation, decreases viscosity and are improved low-temperature properties.

According to the results of numerous studies, was revealed a decrease in the toxicity index of the exhaust gases of the engine, while operating with the DCF. This becomes very important at the present time.

2. RESEARCH OF WORK INDICATORS OF TRACTOR DIESEL ENGINE.

For using DCF on the basis of bioethanol and ricinic oil produced from castor oil plant, was made a study of the effect of the constituent components on the economic, power performance of the diesel engine operating cycle.

The parameters of the D-240 engine performance were studied using 7 mixtures with different component ratios: diesel gas oil (DO) + ricinic oil (RicO) + alcohol (Al), respectively (mixture 1, mixture 2, mixture 3, mixture 4, mixture 5, mixture 6, mixture 7) according to Prokopenko R. M., Horosh A.I., Bashirova R.M. [9,10,11] presented in Table 3.

For calculation, it is necessary to determine the content of the basic substances in ricinic oil according to the method of Egazaryants C.B. [12] and calculate the elementary composition of the mixture, which is determined by the content of C, H, O by the Mendeleev D.I. formula and compared with similar mixtures studied without the addition of bioethanol.

Table No. 2- Elementary composition of mixtures.

№	Indicators	Carbon C	Hydrogen H	Oxygen O
1.	DO444	0,870	0,126	0,004
2.	Mixture 1 RicO+DO	0,8495151	0,1239453	0,0265396
3.	Mixture 2 RicO+DO	0,8426868	0,1232604	0,0340528
4.	Mixture 3 RicO+DO	0,8358585	0,1225755	0,041566
5.	Mixture 1	0,7032584	0,1253704	0,1713712
6.	Mixture 2	0,7244589	0,1234731	0,152068
7.	Mixture 3	0,74871575	0,12389425	0,12739
8.	Mixture 4	0,7517721	0,1262127	0,1220152
9.	Mixture 5	0,76237235	0,12526405	0,1123636
10.	Mixture 6	0,7729726	0,1243154	0,102712
11.	Mixture 7	0,8214863	0,1251577	0,053356

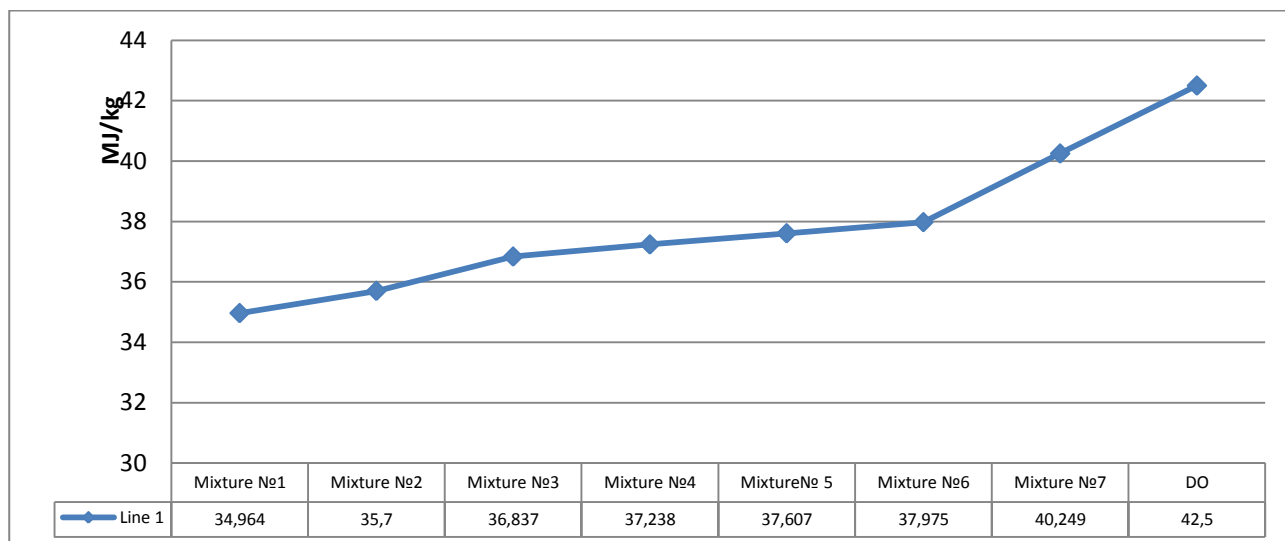
From the obtained data, it can be seen that the addition of bioethanol increases the amount of oxygen and hydrogen in the elemental composition of the mixture. The study of the mixture

viscosity revealed that it is possible to increase the proportion of ricinic oil without adversely affecting the quality of the spray by adding bioethanol.

Table № 3 - Indicators of the working cycle of a diesel engine using mixtures of different concentrations.

№	Indicators	DO	Mixture 1	Mixture 2	Mixture 3	Mixture 4	Mixture 5	Mixture 6	Mixture 7
	1	2	3	4	5	6	7	8	9
1.	Heat of fuel combustion, MJ / kg	42,5	34,964	35,702	36,837	37,238	37,607	37,975	40,249
2.	Amount of air, kg \ kg of fuel	14.35	11,68	11,94	12,34	12,48	12,61	12,74	13,54
3.	Index of molecular change	1,041	1,0589	1,0559	1,0531	1,0531	1,0518	1,0505	1,0456
4.	Combustion temperature, °K	2158,0	2149,53	2153,28	2157,4	2156,07	2157,77	2159,3	2165
5.	Mean effective pressure, MPa	0.7233	0,718	0,71	0,716	0,715	0,714	0,714	0,711
6.	Effective coefficient of efficiency	0.363	0,353	0,338	0,352	0,352	0,352	0,352	0,351
7.	Effective flow density	233,3	291,48	282,7	276,86	274,114	271,54	269,04	254,15
8.	Effective power, kW	63,0	62,61	62,48	62,38	62,307	62,245	62,18	61,93
9.	Change in power, %	-	- 0,62%	-0,83%	- 0,98 %	- 1,1%	-1,2%	- 1,302%	-1,7%
10.	Change in specific fuel consumption, %	-	+24,94%	+ 27,68%	+ 18,67%	+ 17,49%	+16,36%	+ 15,32%	+8,94%

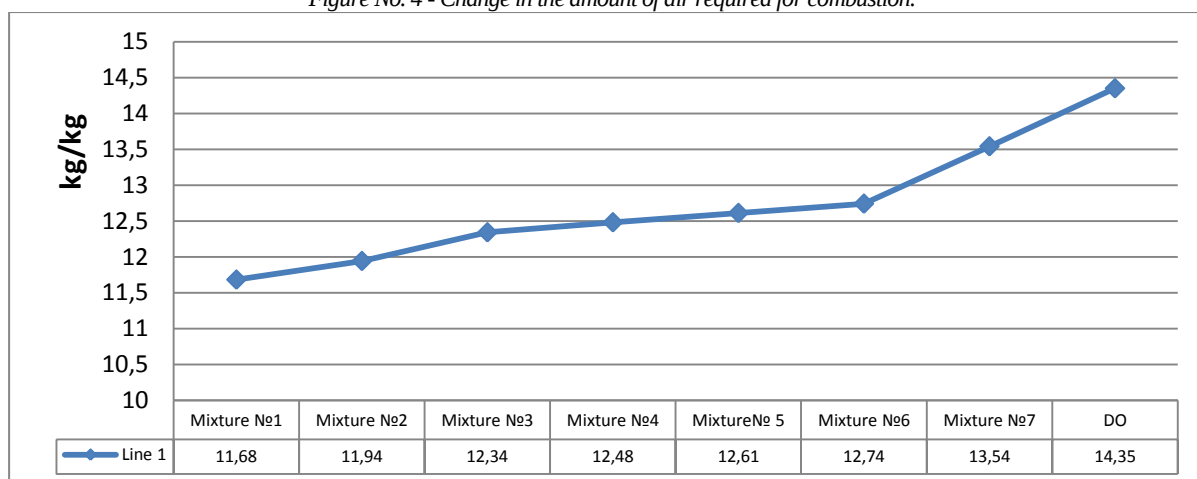
Figure No. 3 - Change in the heat of fuel combustion.



It can be seen that the high content of alcohol significantly influences the reduction of the heat of the fuel combustion, for example, a mixture of 80% DO and 20% RicO has a combustion heat of 40.986 MJ / kg, and a similar mixture with an alcohol of 80x10x10 already has 40.24 MJ / kg. Due to the addition of bioethanol, the

viscosity of the mixture and the low-temperature properties are significantly reduced: the turbidity and crystallization temperature decreases and makes it suitable for use in standard diesel power systems, in terms of quality spraying. [5,8]

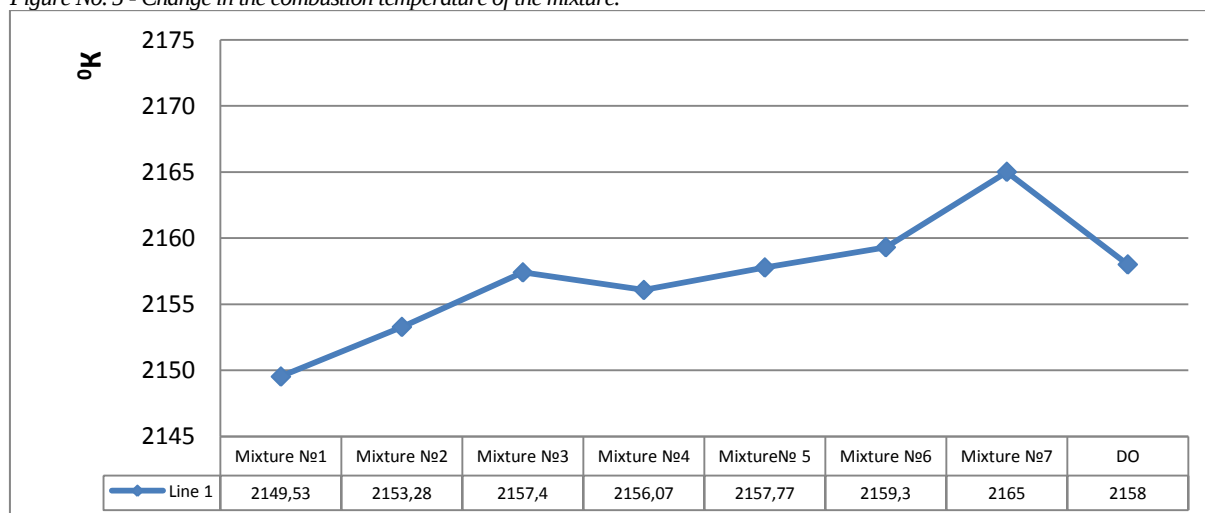
Figure No. 4 - Change in the amount of air required for combustion.



An important feature of the addition of bioethanol in diesel composite fuel mix is the reduction in the required amount of air for firing 1 kg of fuel. From this it can be concluded that for the transfer of the diesel internal combustion engine (ICE) with the system without adjusting the amount of air supplied, it is necessary to modernize the air supply system of the internal - combustion engine.

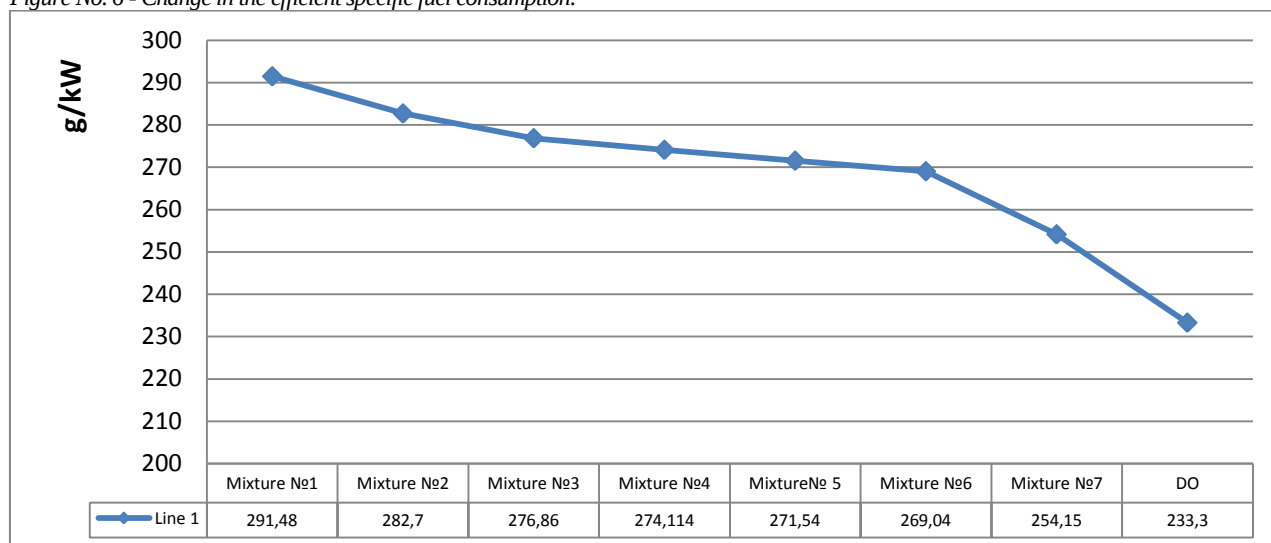
Thus, it is possible to operate the ICE on a depleted fuel-air mixture without modernization, which affects negatively the response of the piston diesel ICE. It is necessary to change the amount of supplied air depending on the added components concentrations.

Figure No. 5 - Change in the combustion temperature of the mixture.



Increasing the proportion of bioethanol also affects the combustion temperature. According to calculations, it was revealed: the lower the amount of diesel fuel is and the higher the amount of alcohol is, the lower the combustion temperature of the mixture is. But with the addition of not more than 25% of the biocomponents, the combustion temperature rises according to the made calculations.

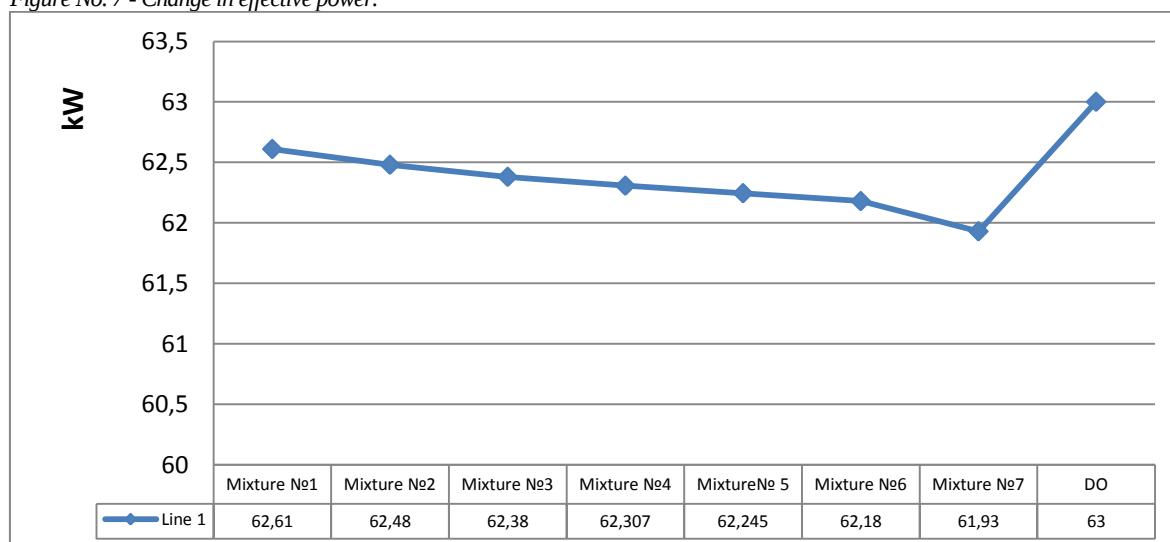
Figure No. 6 - Change in the efficient specific fuel consumption.



According to the schedule, a significant increase in the specific fuel consumption is affected by an increase amount of bioethanol.

If we compare the mixture with the 60% content of DO, it can be seen that the specific consumption increases on average by 2.5 g / kW with a change in the alcohol content for every 5%.

Figure No. 7 - Change in effective power.



But the presence of alcohol favorably influences the performance of ICE, so based on the obtained data on the example of mixtures with a 60% content of DO, it is evident that when the alcohol content is changed by 5%, the effective power increases on the 62-70 W.

3. CONCLUSIONS

1. As a result of reducing the required amount of air for combustion of mixture, occurs a significant change in the α coefficient - a coefficient of air excess;
2. The addition of bioethanol makes it possible to increase the proportion of ricinic oil in the mixture without adversely affecting the quality of the spray;
3. The increased share of biocomponents allows increasing the autonomy of the enterprise and reducing the cost of products;
4. The specific consumption increases by an average on 2.5 g / kW with a change in alcohol increase by every 5%;

5. With an increase in the content of alcohol for every 5%, the effective power increases by an average on 62-70 W, compared to similar proportions of RicO and DO without the addition of alcohol.

6. It was found that the lower the amount of diesel gas oil is and the higher the amount of alcohol is, the lower the combustion temperature of the mixture is. But with the addition of not more than 25% of the biocomponents, the combustion temperature rises according to the performed calculations.

4. CONSEQUENCE.

A multicomponent diesel composite fuel made from castor oil plant and traditional petroleum feedstocks can be used to solve the problems of oil fuel substitution. It is more advantageous to use it at enterprises situated far from large settlements, in connection with which it is possible to achieve a reduction in the cost of production, provided that the components are produced independently or in cooperation with other producers.

5. CONFLICT OF INTEREST

The authors confirm that the presented data do not contain a conflict of interest.

6. GRATITUDES

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THEORETICAL AND EXPERIMENTAL RESEARCH OF AUTOMATIC BALANCING DEVICE

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Abstract: The article deals with the process of wave forming on the surface of a liquid in a chamber of a liquid auto-balancing device partially filled with a liquid for rotors with a vertical axis of rotation under non-stationary modes of the system's motion. In non-stationary processes, the possible wave formation causes a dynamic instability in the operation of the machine and the increase of vibrations under certain operating modes. In the article the problem for the case when the rotor is installed in elastic supports is solved, the joint movements of the rotor - liquid system are considered and their stability is investigated.

Keywords: ROTOR, IMBALANCE, AUTOMATIC BALANCING (SELF-BALANCING), AUTO-BALANCING DEVICE (ABD), PROCESS OF WAVE FORMING ON THE SURFACE OF A LIQUID.

1. Introduction

For machines with variable rotor imbalance and when balancing rotors of machines without stopping under operating conditions, traditional methods of reducing vibrations are ineffective. These machines include separators and centrifuges that are used in various sectors of the national economy (food, chemical, sugar, mining, etc.), medicine, and everyday life. The most reliable, promising, and often the only possible method for reducing the vibrations of such machines is automatic balancing with the help of devices with the free movement of correction masses, which have the form of a hollow chamber partially filled with working bodies (liquid), and are passive direct action regulators that do not require power supply and control system for moving correction masses [1].

The idea fluid balancer is thus not new; but recently there has been a renewed interest, both in industry and in academia [2]. Experimental fluid balancer studies have been carried out by Kasahara [3] and Nakamura [4]. As to mathematical models, simple lumped mass models have been considered by Bae [5], Jung [6], Majewski [7], Chen [8], and Urbiola-Soto and Lopez-Parra [9]. The first and the last two of these papers include experimental studies a swell. The paper [6] includes a few numerical simulation results based on computational fluid dynamics.

To get a fluid balancer to work perfectly is a delicate process and, thus, a more detailed understanding is desirable; in particular, a more detailed understanding of the fluid dynamics of the balancer. This is the main motivation behind the present work.

The characteristic features of the process of automatic balancing of liquid under stationary conditions of rotor movement with a vertical axis of rotation is that automatic fluid balancing is effective for elastically de-formable rotors, on elastic supports ones, where there is a phase difference between the direction of force from the imbalance and the rotor deflection or the displacement of the rotor, as well as the fact that in the auto-balancer device chamber the liquid tends to set against an imbalance not only in the above resonance but also in the pre-resonant area of rotation of the rotor and at the resonance itself [10]. However, there is no theoretical verification of fluid behavior in the ABD chamber under non-stationary conditions of rotor rotation.

Under some simplified assumptions it is [11] shown that the risk of loss of stability of a system containing a cavity partially filled with a liquid is always where the intrinsic frequency of any form of free fluid fluctuations is close to the nutation frequency of the carrier fluid of the body. An infinite number of forms of free fluctuation of a fluid corresponds to an infinite number of areas of instability, however, as shown in [12], only a few first regions can have practical value. It is believed that the internal friction that damps its own fluctuations in the fluid reduces the region of higher order instability.

The purpose of the paper is to analyze the behavior of a fluid in an auto-balancing device chamber for rotors of machines with a vertical axis of rotation in the case of a non-stationary movement of the shaft-fluid ABD system, where it is necessary to take into ac-

count the fluctuations of the free surface of the liquid.

2. Experimental details

In [14] it is shown that the free surface of a fluid is part of a paraboloid of rotation with an axis that coincides with the main central axis of the inertia of the system. As the velocity increases, the parabolic form gets into a cylindrical one. In the horizontal section of the ABD chamber, the free surface of the fluid is a circle. Reducing the radius of the free surface and correspondingly increasing the thickness of the liquid layer at a constant radius of the chamber R does not cause a shift in the centre of the mass of the system, since the centre of the free surface of the liquid coincides with the centre of the mass of the system. In the change of the unbalanced state of the rotary system, a thin layer of fluid is involved, which is close to the magnitude of the double displacement of the centre of the mass of the rotor [10, 13]. Another liquid only increases the mass of the system, located concentrically around the axis of rotation. However, it has been experimentally determined that the amplitude of the system fluctuations increased at a critical frequency range when the ABD chamber was filled with more fluid than necessary for balancing.

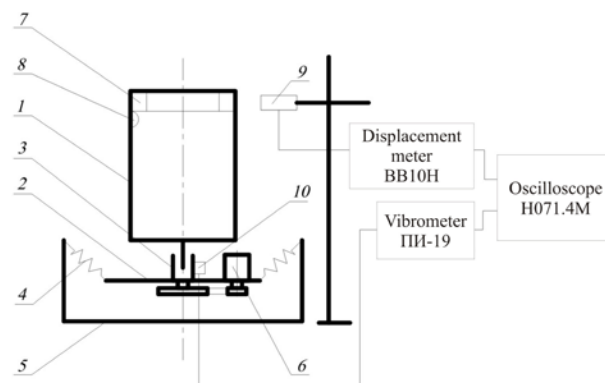


Fig. 1 Installation for studying the behavior of working bodies in the ABD: 1 - drum (rotor); 2 - platform; 3 - bearing housing; 4 - elastic elements; 5 - body; 6 - electric motor; 7 - ABD; 8 - imbalance; 9 - inductive sensor; 10 - piezo sensor (accelerometer) scheme

To investigate fluid behavior in an auto balancing device in the pre-resonance and above resonance areas, an ABD model of optically transparent material was developed and made. The ABD is a ring with an outer diameter $\varnothing 400$ mm, two concentric partitions $\varnothing 300$ and $\varnothing 200$ mm in diameter, forming three concentric chambers for the arrangement of working bodies in them. The installation for studying the behavior of working bodies in the ABD is a hard console vertical rotor (1) elastically suspended to the body (5) (Fig. 1). To ensure sufficient rotor rigidity and the ABD (7) fixed on it, the first is shaped like a hollow cylinder of sheet stainless steel with a bottom at one end, which is rigidly secured to the shaft. The axis of the rotor through the rolling bearings (3) is connected to the rigid platform (2) connected to the body of the unit (5) by four elastically

damping pendants (4) (the actual frequency of oscillations of the suspension system is $\sim 9 - 10$ Hz).

At the free end of the hollow cylinder (further on the rotor), a model of ABD (7) is inserted, which is inserted into the rotor and fixed to it by the outer diameter with the help of clamps. Due to this, the necessary stiffness of fastening of the ABP on the rotor and co-ordination are provided. The ABD model has an inner opening sufficient for free access to the inner cavity of the rotor, which allows, without dismantling the ABP, to change the rotor's imbalance (8) both in location and size by attaching it to the inner wall of the specified load, which during the research additionally holds the centrifugal by forces. The rotor drive is carried out by a synchronous electric motor (6) with a power output of 180 watts through a lowering clutch transmission. The motor is powered by alternating current, whose voltage can vary in a wide range with the help of an autotransformer with a control system. Thanks to this, the installation allows you to set the rotation of the rotor with different frequencies in the range of 0.5-18 Hz (30-1080 rpm), to set different angular accelerations when accelerating the rotor, to stop in the pre-resonance and above resonance areas of rotation of the rotor on a given time for detailed research on the work of the ABD.

In order to determine the effect of the volume of liquid on the efficiency of the system balancing on the resonant rotor frequencies, a series of studies with an imbalance $D = 3000$ g-cm for a variety of radii of the chambers ABD ($R = 0.2, 0.15, 0.1$ m) separately. The chamber stepwise (in 50 ml increments) was filled with liquid (fresh water). After each change in the volume of the liquid, the vibration of the upper edge of the rotor was recorded during its movement. The results of processing records are presented on fig. 2.

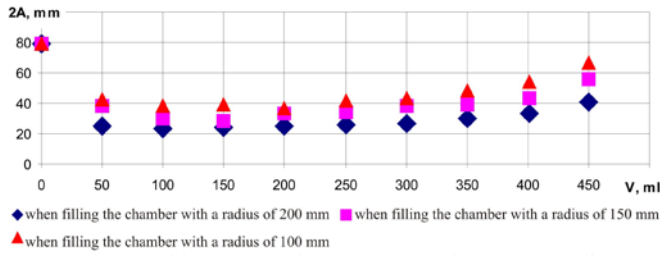


Fig. 2 Amplitude of fluctuations of the upper edge of the drum at different volumes of the working fluid in the ABD

From Fig. 2 it is seen that automatic balancing with liquid working bodies for a vertical rotor at the resonance itself is obvious (comparing with the points corresponding to $V = 0$ ml). There is an optimal amount of fluid for efficient balancing of the rotor, it is 100 ml for a radius of 200 mm, 150 ml for a radius of 150 mm and 200 ml for a camera with a radius of 100 mm with a mass of imbalance of 150 g. In cases where the volume of the working fluid increases, there is an increase in the amplitudes of oscillations.

To explain this behaviour of the system we consider the condi-

$$\frac{\partial u}{\partial t} - 2\omega v = -\frac{1}{\rho} \frac{\partial p}{\partial r} - \ddot{y} \cos(\omega t + \theta) - \ddot{z} \sin(\omega t + \theta) + r\omega^2;$$

$$\frac{\partial v}{\partial t} + 2\omega u = -\frac{1}{\rho r} \frac{\partial p}{\partial \theta} + \ddot{y} \sin(\omega t + \theta) - \ddot{z} \cos(\omega t + \theta),$$

equation of continuity

$$\frac{\partial}{\partial r}(ur) + \frac{\partial v}{\partial \theta} = 0, \quad (2)$$

and expressions for the forces acting on the system

$$M\ddot{z} + c_z z = Rh \int_0^{2\pi} p|_{r=R} \sin(\omega t + \theta) d\theta; \quad (3)$$

tions of wave formation on the free surface of the liquid. To do this, we will analyze the frequency equation of the system of differential equations, which describe the motion of a rotating rotor with an ABD and a liquid inside it.

3. Results of theoretical research

We accept the calculation scheme, shown in Fig. 3. Let us suppose that a simplified flat task is considered (since the geometric dimensions of the ABD have a ratio $R \gg h$ [10]); that the liquid involves into the cylinder chamber of the ABD and rotates with the rotor as a single solid, and that, when neglected by gravitational forces, it has the form of a ring with an internal radius R_1 and external - R . We shall consider the small translational motion of the cylinder in the plane of rotation ($z=z(t)$ and $y=y(t)$ are the generalized coordinates of the cylinder axis). In this case, the movement of the liquid in relation to the cylinder is also considered as small one. In a relative motion, viscosity strengths are not taken into account.

For a moving coordinate system connected with a rotating cylinder, we take the polar coordinate system with the polar axis n (Fig. 3); r, θ - coordinates of a point in this system. In addition, in Fig. 3 the following notation is introduced: $u = u(r, \theta, t)$; $v = v(r, \theta, t)$ - respectively, the radial and tangential components of the relative velocity of the liquid at the point (r, θ) . Denote M - mass of rotor with ABD.

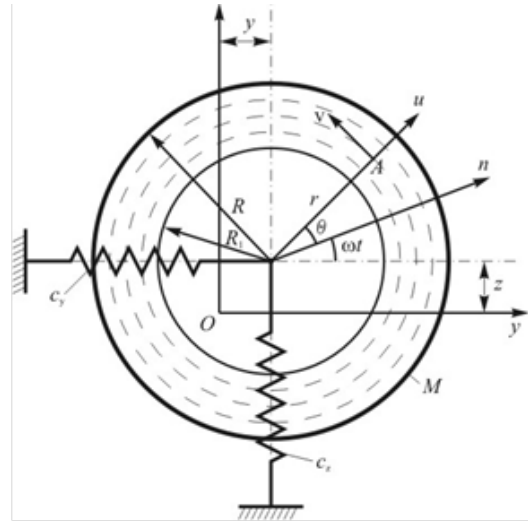


Fig. 3 Calculated scheme

Free oscillations of the rotor-fluid system are described linearized (due to the smallness of the considered motions of the system) by differential equations in the chosen polar coordinate system

$$M\ddot{y} + c_y y = Rh \int_0^{2\pi} p|_{r=R} \cos(\omega t + \theta) d\theta,$$

for limit conditions

$$u|_{r=R} = 0; \quad u|_{r=R_1} = \frac{\partial \zeta}{\partial t}; \quad p|_{r=R_1+\zeta} = 0,$$

where $p=p(r, \theta, t)$ is the liquid pressure at the point (r, θ) , ρ is the density of the liquid; h - height of cylindrical chamber; $\zeta=\zeta(r, \theta)$ is the radial deviation of the points of the free surface of the liquid.

Integration of the system of equations (1) - (3) is experiencing considerable difficulties. Therefore, in view of the study of the sta-

bility of the system, we turn to the analysis of its characteristic equation.

If we look for possible movements of the system in the form of principal oscillations, in which all generalized coordinates change with the same frequency (denote it k), setting the law of the change of the generalized rotor coordinates in the form

$$z=A_1 \sin kt; \quad y=A_2 \cos kt, \quad (4)$$

and take into account only the waves of the 1st order (since in the case of joint oscillations of the system a cylinder - a liquid in a liq-

uid, waves arise only in the first order (the wavelength is equal to the length of the free surface of the liquid) [3]), we find the form of the inner surface of the liquid to the arbitrary constants B_1 and B_2

$$\zeta=B_1 \cos[(k+\omega)t+\theta]+B_2 \cos[(k-\omega)t-\theta]. \quad (5)$$

We find the principal fluctuations of the fluid of the same frequency k .

The solution of the system of equations (1) - (3) with allowance for dependencies (4) and (5) is a system of four equations

$$\left. \begin{aligned} \frac{1}{2}Ak^2 + \left[\frac{2(k+\omega)^2}{(R/R_1)^2 - 1} + (3\omega+k) \cdot (k+\omega) - \omega^2 \right] B_1 &= 0; \\ \frac{1}{2}Bk^2 + \left[-\frac{2(k-\omega)^2}{(R/R_1)^2 - 1} + (3\omega-k) \cdot (k-\omega) + \omega^2 \right] B_2 &= 0; \\ -\left[k^2 - \frac{c_z + c_y}{2(M+m)} \right] B + \frac{c_z - c_y}{2(M+m)} A + \left[\frac{4m(k-\omega)^2}{(R/R_1)^2 - 1} \right] (M+m) B_2 &= 0; \\ \frac{c_z - c_y}{2(M+m)} B - \left[k^2 - \frac{c_z + c_y}{2(M+m)} \right] A - \left[\frac{4m(k+\omega)^2}{(R/R_1)^2 - 1} \right] (M+m) B_1 &= 0, \end{aligned} \right\} \quad (6)$$

where m - mass of fluid in the rotor, $A=A_1 - A_2$, $B=B_1 - B_2$.

In order for system (6) to have a nontrivial solution, its determi-

nant must be zero. From this condition we obtain the equation of frequencies

$$\begin{vmatrix} \frac{1}{2}k^2 & \left[\frac{2(k+\omega)^2}{(R/R_1)^2 - 1} + (3\omega+k) \cdot (k+\omega) - \omega^2 \right] & 0 & 0 \\ \left[k^2 - \frac{c_z + c_y}{2(M+m)} \right] & \left[\frac{4m(k+\omega)^2}{(R/R_1)^2 - 1} \right] (M+m) & -\left[\frac{c_z - c_y}{2(M+m)} \right] & 0 \\ 0 & 0 & \frac{1}{2}k^2 & \left[-\frac{2(k-\omega)^2}{(R/R_1)^2 - 1} + (3\omega-k) \cdot (k-\omega) + \omega^2 \right] \\ \left[\frac{c_z - c_y}{2(M+m)} \right] & 0 & -\left[k^2 - \frac{c_z + c_y}{2(M+m)} \right] & \left[\frac{4m(k-\omega)^2}{(R/R_1)^2 - 1} \right] (M+m) \end{vmatrix} = 0. \quad (7)$$

If $c_z=c_y=c$, then system (6) splits into two unbound systems, and the frequency equation (7) is to two independent expressions, since the left side of (7) for $c_z=c_y=c$ can be given as two multipliers, and the second one is obtained from the first substitute k by $(-k)$. To

solve the question of the stability of the vibrations of the system under consideration for $c_z=c_y=c$ it is enough to investigate one of the equations of the fourth degree, for example

$$\frac{2mk^2(k-\omega)^2}{(M+m)\left[(R/R_1)^2 - 1\right]} + \left(k^2 - \frac{c}{M+m}\right) \cdot \left[-\frac{2(k-\omega)^2}{(R/R_1)^2 - 1} + (3\omega-k) \cdot (k-\omega) + \omega^2 \right] = 0. \quad (8)$$

The division of expression (7) into two independent equations means that for $c_z=c_y$ the principal oscillations of the system are circular. Equations (4) describe the motion of an axis of a cylinder in an elliptical trajectory. However, using the expressions $A=A_1 - A_2$, $B=B_1 - B_2$, an elliptic trajectory can be presented as a sum of two circular, with the angular velocities of these trajectories will be equal to k and $(-k)$

$$z=A_1 \sin kt = \frac{A+B}{2} \sin kt = -\frac{A}{2} \sin(-kt) + \frac{B}{2} \sin kt;$$

$$y=A_2 \cos kt = -\frac{A-B}{2} \cos kt = -\frac{A}{2} \cos(-kt) + \frac{B}{2} \cos kt,$$

which corresponds to the propagation of the wave in the forward and reverse directions (in a moving coordinate system a direct wave propagates towards the cylinder rotation, and the opposite is opposite to the rotation).

An indication of the instability of the system is the presence in the partial equation (8) of the roots with a negative imaginary part. Therefore, if among the four roots of equation (8) there is a couple

of complex-conjugate, motion can be considered unstable.

The roots of equation (8) are the oscillation frequencies of the rotor-liquid system at $c_z=c_y=c$.

Taking the liquid stationary relative to the rotor ($A_1 = A_2$, $B_1 = B_2 = 0$), we obtain the partial frequencies of the propagation of direct and reverse waves and the frequency of oscillations of the rotor-liquid system

$$k_{liq}^+ = \omega \cdot b = \frac{\omega}{1 - \sqrt{\frac{1 - (R_1/R)^2}{2}}};$$

$$k_{liq}^- = \omega \cdot a = \frac{\omega}{1 + \sqrt{\frac{1 - (R_1/R)^2}{2}}}.$$

We express (8) through the partial frequencies

$$ak^2(k-\omega)^2 \cdot (k^2 - k_p^2) \cdot (k - k_{liq}^+) \cdot (k - k_{liq}^-) = 0, \quad (9)$$

where $\alpha = \frac{2m}{M+m} \cdot \frac{1}{1+(R/R_1)^2} < 1$; $k_p^2 = \frac{c}{M+m}$ - the partial

frequency of the rotor with the liquid in it is in a state of rest.

It is shown in [12] that for any α , a , b , k_p^2 , which is the constructive parameters of a system, there always is a rotational speed of the rotor ω , in which the equation (8) will have complex roots, that is, the system will be unstable. For the specific parameters of the system by numerical methods, equating the discriminant equation (9) to zero or using the Euclidean algorithm, one can determine the boundaries of the area of instability.

Thus, the rotating rotor is partially filled with liquid and installed on elastic supports, at some speeds of rotation, it loses its stability. These speeds are not discrete. When $c_x = c_y$ there is one zone of instability, located around the rotational speed ω_{cr} , at which there is a coincidence of the partial frequency of the rotor k_p with a partial velocity of propagation of the return wave in the liquid ($k_{liq}^- = \omega \cdot a$).

With this condition ($k_{liq}^- = \omega \cdot a$) get the formula of critical velocity

$$\omega_{cr} = \sqrt{\frac{c}{M+m}} \cdot \left[1 + \sqrt{\frac{1-(R_1/R)^2}{2}} \right]. \quad (10)$$

Expression (10) defines a central point of an unstable zone. The main parameter characterizing the width of an unstable area is α .

4. Numerical example

The parameter α is a constructive parameter of the rotor-fluid system, which depends on the degree of filling of the ABD chamber with a liquid. In addition, it was numerically checked that an increase in the mass of the fluid leads to the expansion of the instability area. So for such parameters of the system: the weight of the rotor with the camera ABD (M) - 7,3 kg, the weight of the imbalance (m_d) - 0,1 kg, the external radius of the camera ABD (R) - 0,2 m, the height of the chamber ABD (h) - 0,05 (m), mass of liquid (m) - 0,05 - 0,45 kg, liquid density (ρ) - 1000 kg / m³, liquid temperature (t) - 18 °C, critical speed of the rotor (ω_{cr}) - 10 , 14 - 9.64 Hz, the relationship between the degree of filling of the chamber ABD and the coefficient α has the form shown in Fig. 4.

Thus, when the chamber is filled with a liquid weighing 50 g, the instability area of the system is 10,726 - 11,036 Hz (the width of the instability area is 0,31 Hz), and when the 450 g of liquid is filled, the system instability area is 10,047-12,885 Hz (the width of the instability area is 2,839 Hz).

Width of the unstable area, Hz

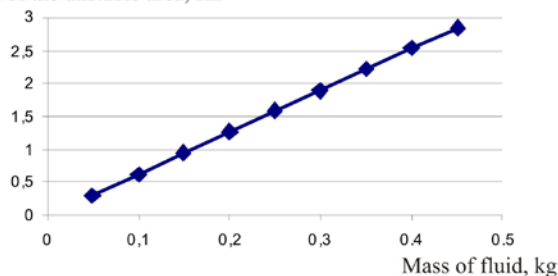


Fig. 4 Increase of the width of the area of motion instability rotor system - ABD - liquid depending on the filling of the ABD chamber with liquid

5. Conclusions

The presence of the areas of instability of the rotor-fluid system and the dependence of the area width on the mass of the fluid filling the chamber of the ABD can be explained by an increase in the amplitudes of the oscillations of the system when the ABP chamber

is filled with more liquid than necessary for balancing. The wave frequency in the fluid resonates with the rotor's rotational speed.

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I. DRACH, A. BUBULIS, A. PAULIUKAS

Theoretical and experimental research of automatic balancing device

INFLUENCE OF THE MODE OF WORK OF MILKING RUBBER ON THE DYNAMICS OF THE STREAM OF TWO-PHASE MIXTURE

ВЛИЯНИЕ РЕЖИМА РАБОТЫ ДОИЛЬНОЙ РЕЗИНЫ НА ДИНАМИКУ ПОТОКА ДВУХФАЗНОЙ СМЕСИ

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Abstract: In the work the characteristic of the change of pressure difference depending on the mode of operation of the milking rubber of the milking machine is given. Graphic model and analytical dependence of pressure change in the inter-wall chamber of the milk glass and the difference of vacuum pressure in the milk hose are developed. On the basis of the analogy of the Euler and Navier-Stokes equations, the equation of the velocity of milk in the milk hose is derived, depending on the vacuum pressure, the intensity of milk flow, the pulsation frequency, the geometric dimensions of the milk hose. The results of the modeling of the speed distribution along the length of the hose and in the interval of the time of one cycle of operation of the milking rubber are given.

Keywords: MILKING RUBBER, INTER-WALL CHAMBER, MILK HOSE, VACUUM PRESSURE, INTENSITY OF MILK FLOW, PULSATION FREQUENCY, VELOCITY OF MILK, MILKING MACHINE.

1. Introduction / Введение

Технологический процесс транспортирования молока доильным аппаратом обусловленный огромным количеством факторов, в частности потерями давления, конструктивными размерами системы, технологическими параметрами работы доильного аппарата, интенсивностью молокоотдачи и др. Особенностью данного процесса есть включения пузырьков воздуха в процессе транспортирования молока в молочном шланге доильного аппарата, что обусловлено конструкцией коллектора и физикой процесса. Очевидно, что включение пузырьков воздуха в молоко приводит к двухфазной среде (коллоидная жидкость – молоко + воздух). Пузырек воздуха в таком физико-химическом состоянии не растворимый в молоке. Поэтому данная смесь считается двухфазной.

Режим транспортирования молока доильным аппаратом адекватный режиму работы доильной резины и соответственно молокоотдачи коровы. Такое движение есть пульсирующее с частотой пульсации, равной частоте работы доильного аппарата, соотношение молока и воздуха изменяется в процессе доения и характер увеличения и уменьшения скорости транспортирования молочной смеси нестабильный.

Учитывая, что значение физических параметров двух компонентов (молока и воздуха) разные и режимы протока их в отношении к ординате молочного шланга (ось шланга) будут отличаться, можно утверждать, что модель технологического процесса с распределенными параметрами есть уравнение гидромеханики.

Исследованиями движения двухфазной среды, в частности смеси воздуха и воды, пара и воды занимались Кутателадзе С.С. и Накоряков В.Е. [1]. Ними разработаны экспериментальные модели потока и теплообмена двухфазных сред при избыточном давлении в жестко закрепленных трубах за отсутствия фактора пульсации потока среды. При моделировании режимов потока среды принимали, что газожидкостная пузырьковая смесь движется при малых числах Рейнольдса i низком газосодержании [2; 3]. Процесс движения происходит при избыточном давлении с подводом энергии, в виде тепла, к смеси.

James P. Brill и H. Dale Beggs [4], а также Paulo A.B. de Sampaio, Jose' L.H. Faccini и Jian Su [5] рассматривают как баланс импульсу для принятого участка молокопровода с двухфазным потоком молоковоздушной смеси и представляют в виде уравнения градиента давления. При этом не учитывают принудительный пульсирующий характер движения. Также ряд исследователей твердят что градиент давления состоит из

гравитационных сил (R. Balasubramaniam, E. Ramé, J. Kizito, and M. Kassemi [6]), сил инерции (Valente Hernandez Perez [7]) и сил трения двухфазной среды об трубу (Nerea Herreras Omagogeascoa [8]), при этом не учитывают увеличение и уменьшение скорости при импульсном характере начала и окончания движения смеси с частотой импульса 1 Гц. В частности, Чен Н. Х. утверждает, что коэффициент трения есть функция числа Рейнольдса с градацией до 2000 и до 4000 [9].

2. Preconditions and means for resolving the problem / Предпосылки и средства для решение проблемы

Анализ исследований изменения вакуума в молочном шланге [10] показал, что характер изменения давления связан с частотой пульсаций доильного аппарата.

Поэтому для моделирования характера потерь вакуума принимаем следующую имитационную модель. При такте сосания (от 0 до 0,52 секунды) присутствуют минимальные потери вакуума ($\Delta p \approx 0$), которые условно равны 1 (единице). На этом участке времени будет "прямая линия". На отрезке времени от 0,52 до 0,72 секунды (переход от такта сосания) потери вакуума будут увеличиваться до номинального значения Δp , от 0,8 до 1,0 секунды потери вакуума уменьшатся до минимального значения.

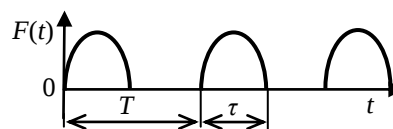
Графическая модель характера изменения потерь вакууму на участке времени $\tau = 0,52-1,0$ с описывается следующим уравнением:

$$(1) \quad f(\tau) = 7,9093 - 20,13 \cdot \tau + 13,088 \cdot \tau^2.$$

Повторяемость потерь вакуумметрического давления Δp зададим графически (фиг. 1), которую аналитически опишем аналогично [11], следующим уравнением:

$$(2) \quad F(t) = \begin{cases} 1, & n \cdot T < t < n \cdot T + \tau \\ \Delta p \cdot f(\tau), & n \cdot T + \tau < t < (n+1) \cdot T \end{cases},$$

где $n = [\tau/T]$ – целая часть числа τ/T .



Фиг. 1 Периодичность перепада вакуумметрического давления Δp : τ – интервал времени присутствия перепада давления; T – период повторяемости

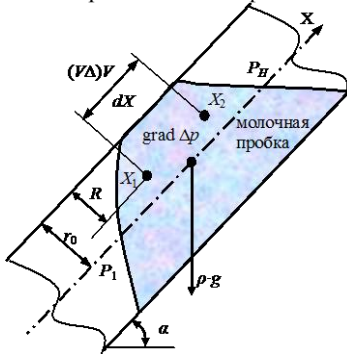
Одним из основных параметров, влияющих на динамику потока есть скорость, которая зависит от многих факторов, в частности от интенсивности молокоотдачи, вакуумметрического давления, линейных размеров молочного шланга, характеристики среды, режима работы доильного аппарата. Учитывая что работу доильного аппарата регламентирует пульсатор, который обеспечивает изменение давления от вакуума к атмосфере и наоборот, этот фактор предопределяет движение молока в соответствии работе пульсатора. Движение молока обуславливается перепадом давлений, который образуется за счет подачи воздуха в коллектор. В результате этого происходит движение двухфазной среды (молоко + воздух).

Рассмотрим модель состояния движущей среды при помощи функций, которые определяют распределение скоростей смеси в продольном и поперечном направлении молочного шланга $v = f(X, Y, Z, t)$ и двух динамических величин, например давления $p = f(X, Y, Z, t)$ и плотности смеси $\rho_{см} = f(X, Y, Z, t)$. Как известно, все динамические величины определяются за значениями двух из них при помощи уравнения состояния среды. Задавая пять величин: три компоненты скорости v , давление p и плотность смеси $\rho_{см}$, полностью определяют движение смеси.

Данные величины есть функциями координат X, Y, Z и времени t . Допустимо, что $v = f(X, Y, Z, t)$ есть скорость смеси в каждой точке X, Y, Z пространства в момент времени t , (относится до конкретных точек пространства, а не к частицам жидкости, которые перемещаются с временем в пространстве). Такие допущения принимаем к величинам $p, \rho_{см}$. Учитывая, что смесь есть молоковоздушная, и при условии, что скорость движения воздуха есть выше в сравнении с скоростью молока, то плотность смеси будет изменяться по длине молочного шланга (по оси X), тогда молоковоздушную смесь будем считать несжимаемой.

3. Theoretical model / Теоретическая модель

Рассмотрим процесс движения молока соответственно с расчетной схемой приведенной на фиг. 2.



Фиг. 2. Схема для определения вектора скорости движения молока в молочном шланге доильного аппарата

Так как жидкость находится в поле тяжести, то на каждую единицу ее объема действует удельная, приведенная к единице объема, сила тяжести $\rho \cdot g$. Поэтому уравнение движения жидкости будет подано в виде уравнения Эйлера

$$(3) \quad \frac{dV}{dt} + (V\Delta)V = -\frac{1}{\rho}(\text{grad}\Delta P - \rho \cdot g)$$

где $\frac{dV}{dt}$ - проекция ускорения;

$(V\Delta)V$ - разница скоростей (в один и той самый момент времени) в двух точках, разделенных расстоянием dX , пройденным рассматриваемой частицей за время dt ;

$\text{grad}\Delta P$ - градиент давления;

$\rho \cdot g$ - составляющая силы тяжести.

Можно допустить что: профиль распределения вектора скорости не зависит от продольной координаты X ; от координаты X зависит составляющая силы, которая действует на частицу жидкости; проекция скорости на ось Y и Z будет равна нулю, а проекция скорости на ось X будет зависеть от X, Y, Z ; жидкость несжимаемая с постоянными параметрами v и ρ .

Тогда уравнение неразрывности удовлетворяется тождественно, а Y и Z - компоненты уравнения Навье-Стокса дают

$$(4) \quad \frac{dP}{dy} = \frac{dP}{dz} = 0,$$

то есть давление постоянно в поперечном сечении шланга.

X - компонента уравнения (3) даст

$$(5) \quad v \cdot \left(\frac{d^2V}{dy^2} + \frac{d^2V}{dz^2} \right) = -\frac{1}{\rho} \left(\frac{d\Delta p}{dx} - \rho \cdot g \cdot \sin(\alpha) \right).$$

Для решения уравнения (5) зададимся начальными условиями: $V = 0$ на контуре сечения шланга; выберем начало координат в центре поперечного сечения шланга и введем цилиндрическую систему координат. Тогда уравнение неразрывности (5) примет вид:

$$(6) \quad \frac{1}{r} \cdot \frac{d}{dr} \left(r \cdot \frac{dV}{dr} \right) + \frac{1}{r} \cdot \frac{dV}{d\varphi} = -\frac{1}{v \cdot \rho} \cdot \left(\frac{d\Delta p}{dx} + \rho \cdot g \cdot \sin(\alpha) \right).$$

В силу симметрии имеем:

$$(7) \quad V = V(r) \rightarrow \frac{dV}{d\varphi} = 0.$$

Уравнение (6), учитывая уравнение (7), примет следующий вид

$$(8) \quad \frac{1}{r} \cdot \frac{d}{dr} \left(r \cdot \frac{dV}{dr} \right) = -\frac{1}{v \cdot \rho} \cdot \left(\frac{d\Delta p}{dx} - \rho \cdot g \cdot \sin(\pm \alpha) \right).$$

Уравнение (8) напомним в следующем виде

$$(9) \quad \frac{1}{r} \cdot \frac{d}{dr} \left(r \cdot \frac{dV}{dr} \right) = -\frac{1}{v \cdot \rho} \cdot \left(\frac{d\Delta p}{dx} - \rho \cdot g \cdot \sin(\pm \alpha) \right),$$

и после последовательного двойного интегрирования по dr получим следующее уравнение, заменив r на R

$$(10) \quad V = \left[-\frac{1}{v \cdot \rho} \cdot \left(\frac{d\Delta p}{dx} - \rho \cdot g \cdot \sin(\pm \alpha) \right) \right] \cdot \frac{R^2}{4} + C_1 \cdot \ln R + C_2.$$

Постоянные C_1 и C_2 определяем с граничных условий. Для молочного шланга круглого сечения радиусом r_0 имеем: при $R = r_0, V = 0$; при $R = 0, V$ - конечная величина. Подставив $R = 0$ в уравнение (10), получим при $C_1 = 0$ бесконечное значение скорости в центре. Соответственно для удовлетворения другого граничного условия необходимо принять что $C_1 = 0$. Используя первое граничное условие, находим:

$$(11) \quad C_2 = \left[\frac{1}{v \cdot \rho} \cdot \left(\frac{d\Delta p}{dx} - \rho \cdot g \cdot \sin(\pm \alpha) \right) \right] \cdot \frac{r_0^2}{4}.$$

Подставим значение C_2 в уравнение (10) получим уравнение параболического распределения вектора скоростей в поперечном сечении молочного шланга

$$(12) \quad V = \left[\frac{1}{v \cdot \rho} \cdot \left(\frac{d\Delta p}{dx} - \rho \cdot g \cdot \sin(\pm \alpha) \right) \right] \cdot \left(\frac{r_0^2 - R^2}{4} \right).$$

где $\frac{d\Delta p}{dx}$ - градиент давления;

$\frac{d\Delta p}{dx}$

v - кинематическая вязкость молока, $\text{м}^2/\text{с}$;

ρ - плотность молока, $\text{кг}/\text{м}^3$;

g - ускорение свободного падения, $\text{м}/\text{с}^2$;

α - кут наклона молочного шланга до горизонтальной плоскости, град.;

r_0 - внутренний радиус молочного шланга, м.

R - радиус от центра до точки определения скорости, м;

После последующего интегрирования уравнения (12) по dx , получим:

$$(13) \quad U_{\text{л}} = \frac{1}{4 \cdot \eta} [\Delta p - X \cdot \rho \cdot g \cdot \sin(\pm \alpha)] \cdot \frac{r_0^2 - R^2}{X} + B.$$

Постоянную интегрирования B определяем с граничных условий: пусть $X = 0$, тогда при $\Delta p = 0$, $V = 0$. Подставим эти значения в уравнение (13) и получим, что $B = 0$. Уравнение (13) запишем в виде:

$$(14) \quad U_{\text{л}} = \frac{1}{4 \cdot \eta} \cdot [\Delta p - X \cdot \rho \cdot g \cdot \sin(\pm \alpha)] \cdot \frac{r_0^2 - R^2}{X},$$

где $U_{\text{л}}$ - скорость движения молока, м/с;

η - динамическая вязкость молока, Па · с;

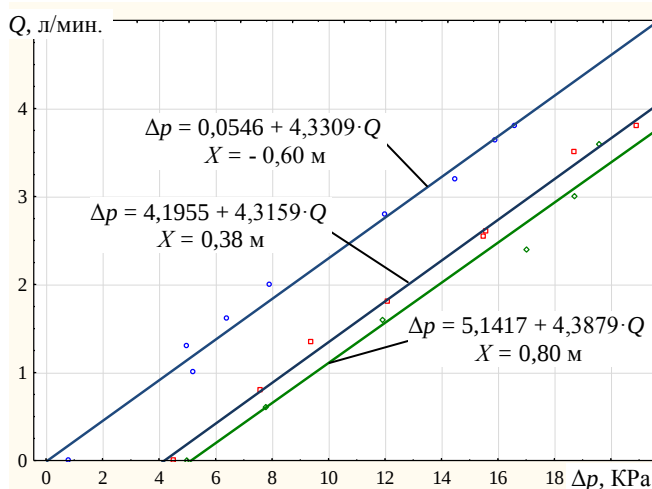
Δp - перепад вакуумметрического давления в молочном шланге, Па;

X - разница геометрических высот начала (коллектор доильного аппарата) и точки определения скорости в молочном шланге, м;

$X \cdot \rho \cdot g \cdot \sin(\pm \alpha)$ - потери вакуумметрического давления, кПа.

4. Modeling results / Результаты моделирования

За экспериментальными исследованиями изменения перепада вакуумметрического давления в молочном шланге доильного аппарата построен график (фиг. 3). При разнице геометрических высот $X = 0,8$ м перепад вакуумметрического давления изменяется по линейной зависимости от 1 до 22 кПа при интенсивности молокоотдачи соответственно от 0,2 до 4,0 л/мин. При $X = 0,4$ м перепад вакуумметрического давления (при тех самых значениях интенсивности молокоотдачи) изменяется от 0,7 до 20 кПа.



Фиг. 3 Экспериментальная зависимость перепада вакуумметрического давления Δp в молочном шланге доильного аппарата от интенсивности молокоотдачи Q при разнице геометрических высот X начала и конца молочного шланга

Моделирование процесса движения молоковоздушной смеси проводили в программной оболочке "Pascal". Динамика процесса транспортирования смеси зависит от динамики изменения перепада вакуумметрического давления на протяжении одного цикла работы доильного аппарата. Для моделирования динамической скорости движения молоковоздушной смеси в молочном шланге запишем зависимость (15):

$$(15) \quad U_{\text{л}}(t_j) = \sum_{i=1}^{i=n} \frac{U_{\text{л}}(r_i F(t_j))}{n},$$

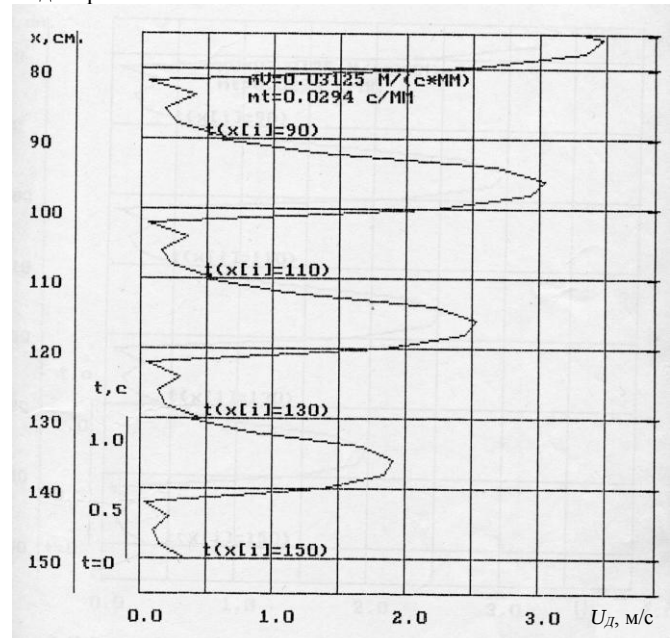
где $U_{\text{л}}(t_j)$ - усредненное значение по радиусу поперечного сечения молочного шланга динамической скорости смеси в t_j момент времени цикла работы пульсатора доильного аппарата;

$U_{\text{л}}(r_i F(t_j))$ - значение скорости в i -й точке радиуса поперечного сечения при значении перепада вакуумметрического давления в j -й момент времени цикла работы доильной резины (уравнение (14));

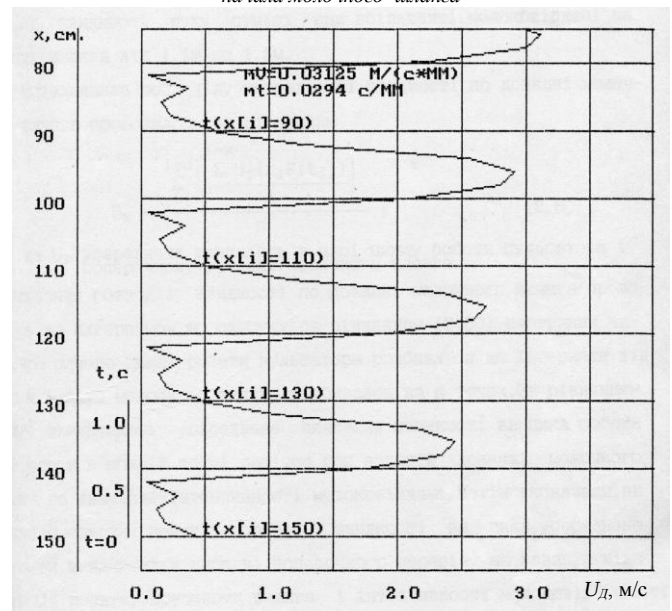
$F(t_j)$ - динамическая функция изменения перепада вакуумметрического давления на протяжении одного цикла работы доильной резины в относительных единицах (уравнение (2));

t_j - момент времени цикла работы доильной резины в интервале от 0,0 до 1,0 секунды.

На фиг. 4 и 5 приведены графики результатов моделирования распределения усредненной по радиусу поперечного сечения динамической скорости при интенсивностях молокоотдачи 0,5 и 4,0 л/мин. и в плоскостях поперечного сечения 0,9, 1,1, 1,3 и 1,5 метра от начала молочного шланга, как одного с вариантов результатов моделирования.



Фиг. 4 Зависимость динамической скорости $U_{\text{л}}$ от времени при интенсивности молокоотдачи $Q = 0,5$ л/мин. и расстоянии $X[i]$ от начала молочного шланга



Фиг. 5 Зависимость динамической скорости $U_{\text{л}}$ от времени при интенсивности молокоотдачи $Q = 4,0$ л/мин. и расстоянии $X[i]$ от начала молочного шланга

4. Conclusion / Заключение

При такте сосания перепад вакуумметрического давления в молочном шланге доильного аппарата практически равен нулю, и скорость транспортирования молока остается

минимальной и удерживается на протяжении всего такта сосания. Перепад вакуумметрического давления происходит за счет прихода молока с сосков вымени и подсоса воздуха в коллектор. При переходе к такту сжатия, по мере убывания вакуумметрического давления в межстенной камере доильных стаканов, увеличивается величина перепада вакуумметрического давления в молочном шланге доильного аппарата. Характер изменения вакуумметрического давления в коллекторе идентичен при разных интенсивностях молокоотдачи. Это объясняется поступлением воздуха в коллектор доильного аппарата.

В начальный момент времени (при $t_j = 0-0,54$ с) в такте сосания транспортирование смеси практически отсутствует. В такте сжатия наступает процесс транспортирования. Скорость увеличивается до максимального значения при $t = 0,7$ с, а потом происходит снижение скорости (фиг. 4 и 5). При этом проявляется тенденция снижения максимальной скорости при увеличении молокоотдачи на длине молочного шланга до 1,1 м и последующее нарастание максимальной скорости транспортирования молока при увеличении молокоотдачи на длине молочного шланга от 1,1 до 1,5 м.

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INVESTIGATION OF DISTRIBUTION OF SOLUTION OF AIR-INJECTOR NOZZLES WITH ADDITIONAL COMPRESSED AIR

ИЗСЛЕДВАНЕ НА ВЪЗДУШНО-ИНЖЕКТОРНИ ДЮЗИ С ДОПЪЛНИТЕЛНО ПРОДУХВАНЕ С ВЪЗДУХ

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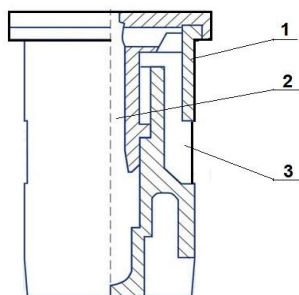
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Abstract: The paper presents investigation of distribution of solution of flat spray nozzles work with additional compressed air and change them function of distribution.

Keywords: air injector nozzles, additional compressed air.

1. Въведение

Въздушно-инжекторните дюзи с въздушна аспирация [3] позволяват пръскане с въздушно-водна емулсия, което според производителите им води до повишаване на масата на емулсионните мехурчета, а от там и до повишаване на кинетичната им енергия. Аериращия ефект на дюзата се осъществява при преминаване на работния разтвор през дифузора - 2 и, при което се понижава налягането на потока на разтвора и засмукване на въздух през специално изработените за целта отвори - 3 (фиг. 1).



Фиг. 1. Устройство на въздушно-инжекторна дюза.

1-тяло на дюзата; 2- дифузор; 3-аериращ отвор.

Възможност за разширяване на състава на емулсията и разхода му дава допълнителното продухване със сгъстен въздух [2] .

Целта на разработката е да се изследват характеристиките на разпределение на разтвора на въздушно-инжекторни дюзи при допълнително им продухване с въздух.

2. Материал и метод

Характеристиките на разпределение на разтвора на дюзите се снимат на специално разработен за целта стенд в катедра „Механизация на земеделието“ при Аграрен Университет – Пловдив (фиг. 2) [1] , като каналите на събирателната му повърхност са през 50 mm.

Допълнителното продухване с въздух на дюзите се осъществява посредством поставен върху дюзата пръстен-2 (фиг. 2). Чрез него сгъстения от бутален компресор въздух се подава към страничните отвори на дюзата.

Измерванията са проведени с процепни въздушно-инжекторни дюзи и №4 с три наляганения на работната течност – p_p

$p_{p1}=0,1$ MPa; $p_{p2}=0,15$ MPa и $p_{p3}=0,2$ MPa

Налягането на работната течност се задава и автоматично се поддържа от стенда (фиг. 1).

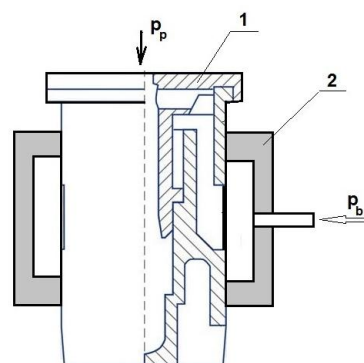
Налягането на допълнителния въздух – p_b е поддържано на четири нива:

I ниво – без аспирация, т.е. отворите на дюзите за въздух са затворени.

II ниво – с аспирация, отворите са отворени и в дюзата свободно се засмуква въздух.

III ниво – налягане на допълнителния въздух $p_b=0,1$ MPa

IV ниво - налягане на допълнителния въздух $p_b=0,2$ MPa



Фиг. 2. Конструкция на дюза за допълнително продухване с въздух.

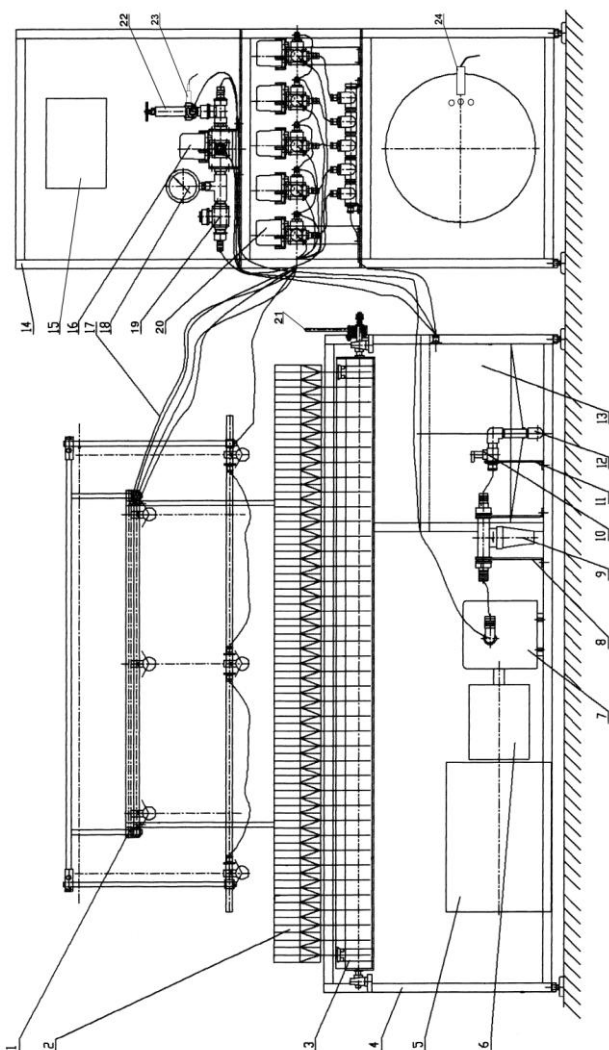
1- въздушно аерирана дюза; 2 – пръстен за подаване на въздух под налягане.

Опитите са проведени като е спазвано условието винаги налягането на разтвора p_p да е по-голямо с поне 0,05 MPa от налягането на въздуха.

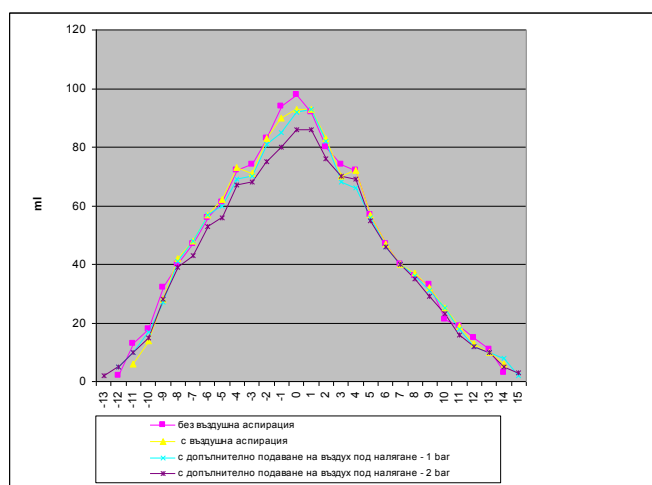
Допълнителния въздух се подава от едноцилиндров бутален компресор с електрическо задвижване, като налягането му се задава и поддържа чрез монтиран към компресора пресостат.

Изпитваните дюзи са поставени на височина 0,5m от събирателната повърхност. Количеството на разтвора се отчита в продължение на 1 min работа на разпръсквача.

Разпределението на разтвора в напречно направление на факела на изпръсквания разтвор се определя чрез отчитане на количеството на разтвора от каналите на приемната повърхност на стенда чрез мензури с точност $\pm 0,1$ ml.



Фиг.3 Конструктивна схема на стенд-симулатора: 1 - щангова система с разпръсквачи; 2 - събирателна платформа с канали; 3 - мерителни цилиндри; 4 и 14 - рама; 5 - електродвигател; 6 - съединител; 7 - помпа; 8 и 11 - конзола; 9 - филтър; 10 - спирателен кран; 12 - смукателен тръбопровод; 13 - резервоар; 15 - бордови контролер; 16 - електроуправляем регулатор на налягане; 17 - маркучи; 18 - манометър; 19 - датчик за дебит; 20 - електроуправляеми секционни кранове; 21 - ръкохватка за обръщане на цилиндри; 22 - механичен регулатор на налягане; 23 - датчик за налягане; 24 - датчик за скорост.



Фиг. 4.Разпределение на разтвора при налягане $p_p = 0,25 \text{ MPa}$

Таблица 1: Резултати от изследванията с работно налягане $p_p=0,25\text{MPa}$

Номер на улея	Проточна скорост																													
	ml/min	-13	-12	-11	-10	-9	-8	-7	-6	-5	-4	-3	-2	-1	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Без въздушна аспирация	1290		2	13	18	32	40	47	56	61	72	74	83	94	98	91	80	74	72	57	47	40	36	33	21	19	15	11	3	
С въздушна аспирация	1271			6	14	28	42	48	57	62	73	71	83	90	93	93	83	70	72	57	47	40	37	32	25	19	13	10	6	
Допълнително подаване на въздух - 0,1 MPa	1255		5	10	17	27	41	48	57	60	69	70	81	85	92	93	82	68	66	56	46	40	36	31	25	18	12	10	8	2
Допълнително подаване на въздух - 0,2 MPa	1202	2	5	10	15	28	39	43	53	56	67	68	75	80	86	86	76	70	69	55	46	40	35	29	23	16	12	10	5	3

3. Резултати и анализ

Резултатите от отчитането на количеството разтвор в отделните мензури при $P_r=0,25\text{MPa}$ са дадени в таблица 1.

С резултатите от таблицата са построени графиките на разпределението на разтвора при различни налягания на разтвора и добавъчен въздух (фиг. 4).

С резултатите от Таблица 1 в позиция 0 (центъра на факела) и позиции 13 и -13 е проведен еднофакторен дисперсионен анализ, като за контрола е избран вариантът без аерация. Установена е статистически доказана разлика с вероятност 95%. На тази база може да се заключи, че аерацията и допълнителното подаване на въздух към разпръсквачите води до намаляване на максималната стойност на дебита и води до разширяване на обхвата на факела.

С резултатите за сумарния минутен дебит (таблица 1) не е установена доказана статистическа разлика т.е. аерацията и допълнителното подаване на въздух не променят минутния дебит на разпръсквачите.

4. Изводи

1. Допълнителното продухване със сгъстен въздух на въздушно-инжекторните разпръсквачи намалява максималния дебит в характеристиката на разпръсквача и разширява обхвата на факела му.

2. Допълнителното продухване не води до промяна на минутния дебит на разпръсквачите.

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ПЪЛНОТО ПРОИЗВОДСТВЕНО ПОДДЪРЖАНЕ КАТО СРЕДСТВО ЗА ОСИГУРЯВАНЕ КАЧЕСТВОТО НА РЕМОНТИРАНИТЕ МАШИНИ

Total productive maintenance as quality providing tool for repaired machines

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Abstract: The article presents main principles for building a total productive maintenance system as well as their role and structure. An organizational structure of total productive maintenance is determined as well as the essence, functions and tasks of each production unit providing desirable quality of repaired machines.

Keywords: quality, repair of machines, TPM

1. Въведение

Пълното производствено поддържане е стратегия, създадена и развита в Япония, с първоначален фокус върху производството и машините, сега покриваща почти всички дейности във фирмата. Тя е безспорно най-високия връх в управлението на поддържането на оборудването през XX век, взела най-доброто от проактивното поддържане и производственото поддържане на оборудването. Това което характеризира системата като уникална е:

- Последователното включване на всички технологии по поддържане на оборудването в единна взаимно балансирана система, чиято цел е постигане на тотална ефективност на оборудването;
- Масовото обучение на операторите и другите изпълнители на методите за поддържане и обслужване на машините и съоръженията с включването им в тази дейност;
- Цялостно интегриране на операторите на базата на малките групи по усъвършенстване на дейността по поддържането;
- Проектиране на система за поддържане на оборудването по време на целия му жизнен цикъл, както и включване на всички звена в системата;
- Гарантиране на високо равнище на безаварийна работа на машините и оборудването на основата на непрекъснатото усъвършенстване, като основен принцип на системата.

Целта на настоящата работа е да се изясни същността на пълното производствено поддържане и неговата роля за осигуряване качеството на ремонтираните машини.

2. Изложение

Може определено да се каже, че стратегията за пълно производствено поддържане се явява един ключов пример за това какво представлява „Японското управление” с неговите „тотални параметри” на включване на всички сътрудници в усъвършенстване на дейността чрез груповите форми. По своята същност подобно включване означава, че в ход влиза не само физическия и умствения, но също така и творческия потенциал на сътрудниците, в което е и силата на „Японското управление”.

Съвременната фирма представлява високо технологична организация, при която различните машини, апаратура, оборудване и съоръжения играят незаменима роля за постигане на нейните цели. В този смисъл поддържането на оборудването е сред решаващите фактори за постигане на висока производителност, ниска себестойност и ефективност на оборудването. Повредите или влошаване на функционирането на апаратурата, машините и съоръженията въздействат върху дейностите, процесите и организацията чрез следните вреди и последствия:

- Влошаване на качеството на продукцията;
- Нарушаване или забавяне на поръчките;
- Предизвикване на трудови злополуки и аварии, които могат да доведат до значителни щети;
- Отрицателно влияние върху трудовия морал;
- Повишаване на разхода на енергия и материали;
- Замърсяване на работната и околната среда.

Философията на стратегията за пълно производствено поддържане се ръководи преди всичко от виждането за необходи-

мостта от интеграция на всички сътрудници в организационните усилия за кайзен. Изхожда се от това, че там, където няма съзнание, разбиране и воля за промяна и усъвършенстване проблемите не се решават. В тези случаи организацията не просто губи конкурентна способност, но е възможно да се стигне до трагически инциденти и даже смърт в резултат на злополуки.

Целите на пълното производствено поддържане са:

- Създаване на фирмена култура която да максимизира ефективността на производствената система;
- Практично организиране на производството за предотвратяване на загубите преди да са се появили, в рамките на целия жизнен цикъл на производствената система с оглед постигане на нула злополуки, нула дефекти и нула повреди,
- Цялостно ангажиране на всички отдели на фирмата (производство, поддържане, проучване и развити, маркетинг и др.);
- Ангажиране на всички от висшият мениджмънт до машинните оператори;
- Постигане на нула загуби чрез дейности в малките групи.

Пълното производствено поддържане се основава на следните осем принципа:

- Автономно поддържане;
- Планирано поддържане;
- Фокусирано подобрене;
- Образование и обучение;
- Поддържане на качеството;
- Контрол на подобренията;
- Безопасност, здраве и околна среда;
- Офис на пълното производствено поддържане.

Автономно поддържане представлява обучение на работниците как да поддържат машините с които работят чрез проверки, почистване, смазване, инспектиране и притягане, за ранно откриване на проблемите по машините. Те предотвратяват ускореното износване на оборудването, коригират аномалиите по машините, елиминират престойте поради малки повреди, дефекти и други загуби. За отдела по поддържане остават само задачи изискващи специални умения, при решаването на по-сложни проблеми и когато работата крие риск от злополуки.

Планираното поддържане ангажира поддържащия персонал в дейности, целящи увеличаване на ефективния фонд от работно време на машините. Освободени от малките ежедневни проблеми, те се фокусират върху системата за превантивно поддържане и управление на резервните части, както и в реализиране на кайзен (подобрения по машините).

Фокусираното подобрене насочва всички дейности в пълното производствено поддържане по отношение на откриване и намаляване на загубите в производствената системата. Загубите според пълното производствено поддържане са: престой и накъсване на работата, понижена производителност, дефекти в продуктите, излишъци на материали и незавършено производство, излишен транспорт и пренасяне, загуби на енергия и материали.

Образованието и обучението цели непрекъснато развитие на персонала и в частност работниците, чрез осигуряване на адекватно обучение и оценяване на уменията.

Поддръжане на качеството е насочена към определяне, стандартизиране и фиксиране на условията за запазване на качеството чрез поддръжане на машините, методите и материалите, с цел постигане на продукти без проблеми с качеството.

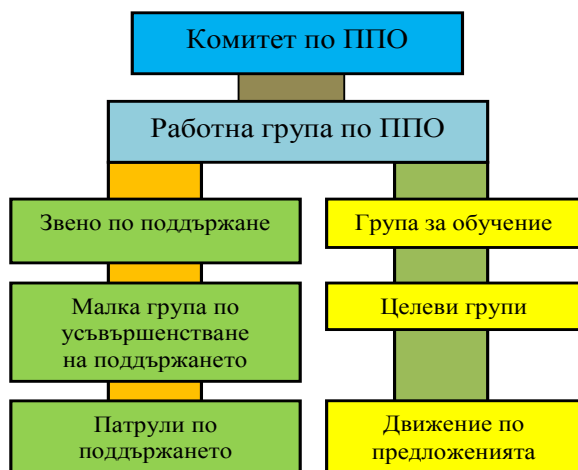
Контрол на подобренията включва анализиране и документирание на направените подобрения и отстранените проблеми в оборудването и процесите, за да може тази информация да се използва при създаване и внедряване на новите машини, процеси или продукти.

Безопасност, здраве и околна среда създава система за безопасна и здравословна работна среда, за да се постигнат нула злополуки.

Офиса на пълното производствено поддръжане цели административните функции да бъдат високо ефективни, като осигуряват своевременно съдействие и услуги на производствените звена.

Системата **5S** е основа на пълното производствено поддръжане. Тази система поддържа работната среда добре организирана, подредена и чиста като условие за ефективна работа и безпроблемно протичане на всички дейности.

Стимулираща среда за успешното внедряване на пълното производствено поддръжане, представлява изграждането на смесени екипи на всички нива, които да ангажират потенциала на целия персонал. Организационната структура на пълното производствено поддръжане несъмнено е специфична за всяка



Фиг. 1. Структура на пълното производствено поддръжане

фирма и предприятие. Но като правило се прилага един изпитан от многогодишната практика модел, характерен и за други подходи като този на тоталното качество. Организационната структура на пълното производствено поддръжане (фиг.1) включва следните структурни звена и единици:

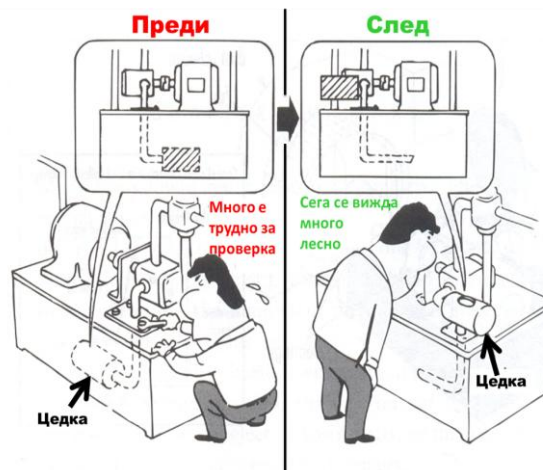
- Комитет за пълно поддръжане на оборудването;
- Работна група по пълно поддръжане на оборудването;
- Група за обучение;
- Звено по поддръжането;
- Малка група по усъвършенстване на поддръжането;
- Целеви групи по поддръжането;
- Патрули по поддръжането;
- Движение по предложенията.

Комитетът за пълно поддръжане на оборудването е под председателството на том мениджър. В състава на комитета влизат мениджърите на основните производствени звена, както и на звената по поддръжане, технологическия отдел, проектно-конструкторското звено, енергомеханичното и производственото планиране. Комитетът взема стратегически управленски решения, свързани с одобрената стратегия за пълното поддръжане на оборудването, осъществява контрол по нейното внедряване, реализация и др.;

Работната група по пълно поддръжане на оборудването е съставена от заместник-ръководители или представители на гореспоменатите структурни звена и оглавявана от заместника на топ мениджъра. Групата разработва и обсъжда цялостната стратегия, програмите, плановите, задачите и мероприятията за внедряване и осъществяване на пълното поддръжане на оборудването, като ги предлага за одобрение от Комитета за пълно поддръжане на оборудването;

Група за обучение по тотално поддръжане на оборудването. Нейната цел е обучение на сътрудниците в организацията на различните методи и техники, свързани с поддръжане на оборудването. За разлика от другите две структури тя има постоянен характер;

Звено по поддръжането, това е щатно звено, което играе важна роля. То е колектив от специалисти, ориентирани да споделят знания и опит по поддръжане на оборудването с всички други звена;



Фиг.2. Осъществяване на Кайзен по машините

Малка група по усъвършенстване на дейностите по поддръжане на оборудването. Тя функционира в рамките на всяко основно производствено звено. Ако има формирана малка група по качеството или по други производствени проблеми, то тя се използва като основа за дейността по поддръжането. На базата на усвоените методи, свързани с поддръжане на оборудването малките групи правят предложения за подобрения по проблемите на обслужването на машините и съоръженията (фиг.2). През 2003 г. само в един от заводите на Toyota в САЩ, са постъпили 80 000 предложения за подобрения по системата кайзен, направени само от екипите в монтажния цех. От тях заводът е внедрил 99 %. На практика тези групи са в основата на системата за инициативи по подобряване на поддръжането. Те са доброволни, но участието в тях се отчита при цялостната оценка на дейността на сътрудника. Оттук и връзката със системите за материално и морално стимулиране, развитие и кариера. Задължително във всяка група като консултант присъства представител на звеното по поддръжане;

Целеви групи по поддръжане на оборудването. Те са съставени от най-добрите специалисти по материалната част на организацията. В определени случаи могат да бъдат привлечени и външни сътрудници - от университети, институти, лаборатории, технически бюра и други. Тези групи имат за задача изпълнение на конкретни задачи, свързани с корективното поддръжане, тоест с дейност по усъвършенстване конструкцията на съществуващите машини и съоръжения;

Патрули по поддръжането, почистването и подреждането. Те се съставят от самите оператори по места. Имат задачата да проверяват регулярно състоянието на работното място, изпълнението на системата 5 S, както и осъществяването на някои основни изисквания, свързани със системата за инициативи за поддръжане. Патрулите играят немалка роля за внедряване на ново съзнание и отношение към оборудването сред сътрудниците;

Движение по предложенията, представлява относително неспециализирана форма, създаваща условия за генериране и внедряване на идеи по усъвършенстване на трудова и организационна безопасност, поддържане на оборудването, начина на функциониране, енергийната и материалната ефективност и други въпроси свързани с оборудването.

3. Заключение

1. Съвременните машини се нуждаят от нов начин на поддържане, който гарантира съкращаване срока на въвеждане в експлоатация, удължаването на жизнения им цикъл и намаляване на постоянните разходи за ремонт. Въвеждането на пълното производствено поддържане може да реши тези проблеми.

2. Установени са основните принципа на изграждане на системата за пълното производствено поддържане. Изяснено е тяхното съдържание и роля в пълното производствено поддържане на машините.

3. Определена е организационната структура на пълното производствено поддържане, а също така същността, функциите и задачите на всяко едно звено.

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DEVELOPMENT OF EQUIPMENT FOR THE STORAGE OF SOYBEANS WITH ACTIVE VENTILATION

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Abstract: The article gives an overview of the creation of technology and equipment for pneumatic drying and storage of soy without attracting external sources of energy, and as storage capacity, use cheap freight containers, which will reduce the energy costs of the process and the material consumption of equipment.

KEYWORDS: SOY STORAGE AND PROCESSING, DRYING, PNEUMATIC CONVEYING, MODULAR CONTAINER EQUIPMENT, AIR FLOW, AIR CYCLONE, GAS-AIR MIXTURE, RECIRCULATING GRAIN, GRAIN MOISTURE CONDUCTIVITY.

1. Introduction

The idea of the labor is to create technology and equipment for active ventilation and storage soy of without attracting external energy sources, and use cheap load containers as storage capacity, which will let to reduce the energy costs of the process and the material consumption of equipment. To solve this problem, a technology is proposed that provides storage of grain in modernized cargo containers of 20, 30, 40 tons installed vertically.

The aim of the labor is to scientifically substantiate and develop container-modular equipment for recycling drying of soy in the course of long-term storage of soy, adapted directly to the conditions of agricultural producers, which is modularly designed to create a modular series of separate parts on demand.

In this case, soybean is loaded and unloaded by pneumatic conveyor, which eliminates leakage and loss of soybean. When stored by the same conveyor, the soy-air mass moves in a closed circle, thereby actively ventilating the mass. This continues until the desired humidity is reached. Drive by electric motor or from tractor power take-off shaft (PTO), which is convenient for remote farms. At the same time, the energy intensity of the drying process and soybean storage is reduced.

2. Prerequisites and facilities for solving the problem

Soy is a unique plant, a miracle of the wild. Now soy is the leading culture of world agriculture, the pinnacle of perfection and universality in the whole plant world. Soy is central place to solve the protein problem and is very profitable. The prospects of both production and consumption are related to the population of the earth, which has a growing tendency. Without the use of soy products, the nutrition structure in many regions will be unbalanced and insufficient. Soy seeds contain 38-42% protein, 18-23 fat, 25-30% carbohydrates, as well as enzymes, vitamins, minerals. Due to the rich and various chemical composition, soy does not know its own equal in terms of production growth rates, it has long been widely used as a universal food, fodder and oilseed culture. It has no analogues in the arsenal of plant resources in terms of productivity and quality composition. Soy oil contains 58% of the structure of vegetable oils (rapeseed, cottonseed, sunflower, palm, etc.). Total world production of soy is about 130 million tons, where the main producers being the USA, China, Argentina, Brazil, the EU countries, accounting for more than 85% of the world market. The share of Argentina, Brazil and the United States in world exports is about 87%. [1,2].

According to the Ministry of Agriculture of the Republic of Kazakhstan in 2016, about 220 thousand tons of soy were produced, of which about 20 thousand tons was exported. The main region of soy production in Kazakhstan is the Almaty region, where the cultivation area is 150,000 ha. In the future, it is planned to implement the investment cluster program "Mozhoko-2020", envisaging an increase in sawing to 400,000 hectares and producing up to 1 million tons of soy a year. At the same time, further growth in soy production is constrained by the lack of a stable alternative market for specific products. In 2016, Kazakhstan sold 10.8 thousand tons to Sweden, 1.4 thousand tons to Kyrgyzstan, 1.6 thousand tons to Tajikistan, and 6.1 thousand tons to Uzbekistan. A

total of about 19.9 thousand tons of 220 thousand produced. Because of this, soy producers are forced to sell to Chinese buyers for a price two times lower than the world price. In the future, China is willing to acquire Kazakhstan soy in unlimited quantities, as they now have a large demand for soy and soy products, which promises to our farmers the optimism on the issue of the future prospects of production of the product [3].

Besides, in Kazakhstan there is an opportunity to supply not only primary products (soy), but processed soy products (meal, vegetable oil and other related.), so there are already three soy-processing factories: JSC Vita-Soya productivity 150 thousand tons per year, LLP Plant on processing of soy Extra-24 thousand tons per year, LLP Sary Bulak-24 thousand tons per year. Unfortunately, they can not reach the planned capacity due to the lack of a stable source of raw materials throughout the year, which implies its storage and preparation under appropriate conditions, since soy with high protein and fat content, as well as increased seed hygroscopic, with unfavorable conditions quickly deteriorates (presence of organic impurities, high humidity). Even dry soy seeds are self-warmed by impurities.

According to the Ministry of Agriculture of the Republic of Kazakhstan, about 2 million tons of grain are stored and processed directly by product producers, i.e. various agricultural formations. At the same time, up to 50% of the volume of this raw material is irrevocable losses. Great danger to the grain is fungi, bacteria, harmful impurity and pathogenic microorganisms carried by pests of cereals - insects, rodents, birds. Toxins formed during the molding process, damaged by insects and mold grains, dirt and odors from pests of grain cause a decrease in its feed value, and animals refuse to eat such grain. Processing and transportation of moldy and damaged grain negatively affect the labor safety and the health of workers. The defeat of the grain by fungi is possible in the field, and during storage. Fighting mycotoxins is a worldwide problem, as in the structure of grain losses, mycotoxins account for 25%. The overwhelming majority of domestic agricultural producers, even in "good" times, did not consider the possibility of building their own elevators. The cost of erecting galvanized silos for storage of grain is estimated at about 150 - 250 dollars per ton. That is, in the equipment for an elevator with a capacity of 20 thousand tons alone it is necessary to invest about 3 to 5 million dollars. And also need to pay for design work, building. The minimum payback period of the elevator is 4-5 years, and this is provided that the agrarian will store on it not only its own grain [4,5,6].

In many areas of the country's climatic conditions predetermine cleaning a large part of the harvest of grain, oilseeds and other crops with high humidity, in which their long-term preservation can not be ensured.

Reduced soy humidity to 12% and bringing crude and wet grain to a stable storage state - the main purpose of drying.

In general, drying is understood as the process of dehydration of materials. This complex process consists of the transfer of heat by heated soy air, the transfer of humidity inside the grain to its

surface, its evaporation in the peripheral layers of each grain, the movement of steam from the peripheral layers of the grain to the surface and into inter granular space, removing it from the mass of the grain. At the same time, a number of physical and biochemical processes that affect soy quality are passing through soya. When drying, the role of air is not limited to the functions of the heat deliverer and humidity absorber. Oxygen of air takes part in biochemical processes in the embryo and endosperm, which increase with increasing temperature. At the same time, the processes of physiological ripening of soy are intensified, which is of great importance for improving its quality. Timely and correctly drying increases the stability of soy during storage, improves its grain and food dignity. Drying accelerates the post-harvest ripening of soy, smoothing the grain mass in terms of maturity and humidity, improves the appearance of soy. Drying has a positive effect on output and quality of soy processing products. Drying acts depressingly on pests and soy micro flora, allows in some cases to improve the technological properties of defective grains (damaged by a bug-tortoise, frostbite, sprouted, etc.) Thus, drying allows to bring soy into a stable state for storage and improve its quality.

In the Institute of AIP and NT KazNAU received a patent for invention No. 31913. Method and device for drying the grain. The essence of the invention lies in the fact that the grain is loaded, dried, cured, dried, cooled and unloaded, characterized in that the permissible moisture level at drying and before drying is limited by the value: $\Delta U_d \leq \text{Kimu. Usp, where Kim.d.}$ - Massive exchangeable permissible number of Kirpichev; n is calculated by: $n = (U_n - U_k) - \Delta U_d / \Delta U_d$, where $U_n, U_k, \Delta U_d$ - initial, final moisture content and permissible moisture content, kg ow / kg dry. mat. . A device for drying grain containing, above the drying bunker, drying chamber, cooling, fan, furnace, loading and unloading way, characterized in that the capacity of the drying chambers is determined from: $G_d = G - n \cdot G_{ot} / \eta_d + n, t$, where G, G is the capacity of the dryer and the vending chamber, t ; η_d, η_d - the fraction of heat entering the evaporation of moisture during drying and drying, h ; n is the number of restraints, in addition, the number of chambers before drying is equal to the number of vending chambers [7].

However, along with this, we propose a method of active soy ventilation. In this case, the material itself moves in the air stream. As a result, cooling and drying of the unmerged material is achieved in order to prevent or eliminate self-heating during mass harvesting, equalize the temperature and humidity of the material during storage, eliminate clumping and form local rotting zones, etc. Along with this, the re-equipped containers are currently used in urban and temporary architecture, for realizing business ideas and creating residential and workplaces. Of the containers build houses, hotels, shops, build shopping centers and even office buildings. The advantages of container construction are obvious - it's cheap, fast, practical and durable. All these positive characteristics of buildings built on the basis of sea containers, could not fail to appreciate the workers of agriculture, agrofirms and private farmers who have found their application of such a universal design, as a metal container block.

Today standard 20-foot and 40-foot containers are actively used as granaries, as well as universal warehouses for agricultural products. Practice has shown that a minimal re-equipment of the container in accordance with the required conditions allows for a long time to preserve the quality and weight of any type of grain.

Analysis of the mechanism of dehydration of the material during convective drying shows that the temperature of the drying agent can be significantly increased to 70 ° C during brief heating of the material. The limit is the temperature at which the evaporation temperature (the temperature of the wet thermometer of the psychrometer) will be equal to or close to the allowable temperature of the grain heating. Experiments show that, at a high temperature of the drying agent, soy warm up to 55 ° C to an

acceptable temperature, and the evaporation of moisture from its surface takes place within a few seconds (about 10 s). Above 55 ° C, the supply of heat to soy above 55 ° C leads to injury to the embryo of the grain [8].

In this connection, we are proposed the most sparing, less energy-consuming method of storage and drying of soy, acceptable for the conditions of direct commodity producers.

3. The solution of the problem under consideration

As a result of preliminary research it was established that for the drying and storage of soy it is most effective and economically expedient to use active ventilation of the material while the material itself is moving in the air stream. As a result, cooling and drying of the material is achieved. As a result, the soy in the tank is stirred, preventing and eliminating self-warming, equalizing the temperature and humidity of the material during storage, excluding caking and the formation of local decay zones, etc.

At the same time, the modernized agricultural containers, agricultural company and private farmers, who have found their application of such a universal design as a metal container block, could not but evaluate the converted containers.

With this in mind, the advantages outlined above can have a greater effect when combined using them.

We offer a new version of container-modular equipment for drying and storage of grain in farms (Fig. 1).

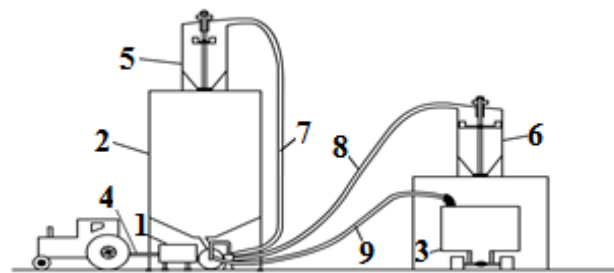


Figure 1- Container-module equipment for drying grain.

1-pneumatic conveyor, 2-equipped container, 3-vehicle for grain storage, 4-VOM tractor, 5,6-cyclone, 7,8,9-hoses for grain loading and unloading

The equipment operates as follows. The process consists of three cycles: loading, venting and unloading.

Loading (Fig. 2) is carried out by means of a pneumatic conveyor (1) from the vehicle (3) through the suction conduit (9), then hoses (7) into the hopper (2), where the grain is stored.

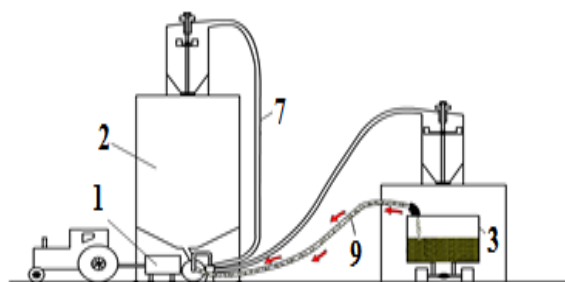


Figure 2-Scheme of grain loading through a pneumatic conveyor

The active ventilation of the grain (Fig. 3) is carried out in a closed circle through the pneumatic conveyor (1) -shoot (7) - cyclone (5) into the storage bin (2).

Ventilation is carried out until the humidity is 12%. At the same time, there is a valve on the pneumatic conveyor to transfer the flow of mass into the corresponding hose.

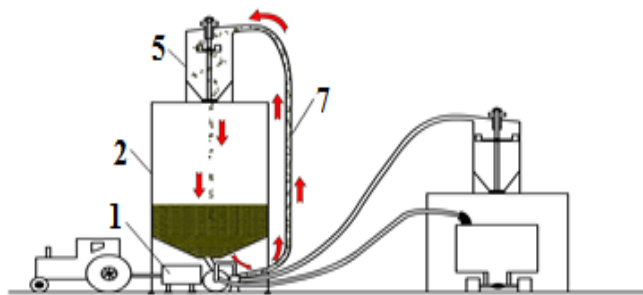


Figure 3- Scheme of ventilation of grain flow in a container

The unloading of grain from the hopper (Fig. 4) comes from the lower part by means of a pneumatic conveyor (1), a hose (8) through a cyclone (6) into a vehicle (3).

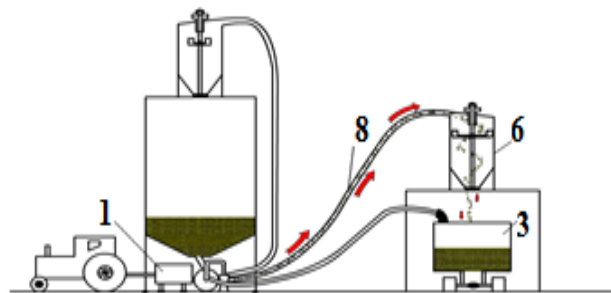


Figure 4. - Grain unloading scheme

Modular equipment of container type are the most advanced devices for rapid cooling, as well as for slow drying of soy and seeds of various crops, intended for the accumulation and temporary conservation of soy, while preserving its sowing and food qualities. The equipment occupies a relatively small space, provides full mechanization of loading and unloading of grain, it is mounted quite quickly. Thanks to the ability to work in any weather, rapid cooling, full mechanization of loading and unloading soy, vented hoppers are suitable for use in combination with inline grain cleaning and drying lines [9].

4. Conclusion

For post-harvest drying of soy, container-modular equipment for pneumatic drying and storage is offered.

The equipment will occupy a relatively small space, provide full mechanization of grain loading and unloading, install quickly enough, and use containers of 20, 40 tons as capacity for storage, which makes it possible to work in any weather, rapid cooling, full mechanization of loading and unloading soya, ventilated bunkers are suitable for use in combination with grain-line drying lines.

The proposed technology of container-modular air drying and storage equipment will allow drying without the use of expensive energy sources (gas, fuel oil, coal). It is necessary to justify the technological constructive scheme of the proposed variant of soy drying, with the conduct of an analytical study. Development of a technical specification for a laboratory sample of a soy drying plant, followed by manufacturing for laboratory testing.

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THEORETICAL INVESTIGATION OF A REAR-MOUNTED LINKAGE FOR WIDE-SPAN TRACTORS

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Abstract: The work is devoted to the investigation of the peculiarities of aggregation of wide-span tractors (vehicles) with agricultural machines and implements. As a result of the conducted studies, it has been established that, from the point of view of the control impact (the turning angle of the driven wheels) and the disturbing impact (the turning angle of the mounted implement in a horizontal plane) exerted by the wide-span tractor, the most desirable is such a variant of configuration of the mounted device in which the instantaneous centre of rotation, formed by the convergence rays of its lower links, is in the centre of the mass of the tractor. When using wide-span tractors (vehicles) with mounted implements of the high 4N and 4 categories in accordance with the ISO 730: 2009, one should consider a possibility to configure them both for a three-point and a two-point scheme.

KEYWORDS: WIDE SPAN TRACTORS (VEHICLES), REAR-MOUNTED LINKAGE, STABILITY AND CONTROLLABILITY, ANGULAR MOBILITY, RESEARCH ON THE PC.

1. Introduction

Wide-span tractors (vehicles) [1, 2] are lately gaining popularity in the world, which allow implementing a controlled traffic farming technology.

Owing to their rather high versatility the wide-span tractors (vehicles) can (and undoubtedly should) be aggregated with mounted, semi-mounted and trailed agricultural machines and implements [1, 2]. At the same time, the wide-span tractor (vehicle), by its layout scheme, is intended for linking of the technological part (with or without a mounted device) in sufficient vicinity to its centre of mass [1]. In such a situation, to ensure that the deviations from the course of the wide-span tractor (vehicle) did not adversely affect the stability of the movement of the mounted aggregated machines and implements, the design of the mounted implement must (to some extent, naturally) be able to independently rotate in a horizontal plane. That is why the issue of justifying the parameters of a wide-span tractor (vehicle) from the standpoint of steady movement of its technological part in a horizontal plane is relevant.

It is known that the three-point adjustment scheme of a rear-mounted mechanism of the conventional tractor can provide sufficient mutual angular turning ability of its framework and the technological part [3]. Considering a possibility to unify the mounted implements of the wide-span tractors (vehicles) with the conventional ones, the parameters of which are determined by the International Standard ISO 730: 2009, it is necessary to establish a correspondence of the sizes and requirements for a three-point mounting, designed for the connection of agricultural machines and implements.

In addition, certain trends have been outlined lately in the improvement of the design of the three-point mounting devices of tractors [4-9]. However, insufficient attention has been paid to the problem of their angular mobility, especially with respect to the wide-span tractors (vehicles) characterised by an untypical layout, the specificity of their use and the operating conditions.

The purpose of the work is to increase the efficiency of operation and use of wide-span tractors (vehicles) by justifying the parameters of their mounted devices.

2. Materials and method

As a physical object of research, we will consider a wide-span vehicle of the authors' design, shown in Fig. 1.



Fig. 1. A design of a wide-span vehicle developed in Ukraine

When making a plane-parallel movement in a horizontal plane, a number of corresponding forces are acting upon the wide-span vehicle and its technological part. For the purpose of mathematical simulation of its plane-parallel movement, we will construct an equivalent calculation scheme for the wide-span vehicle (Fig. 2).

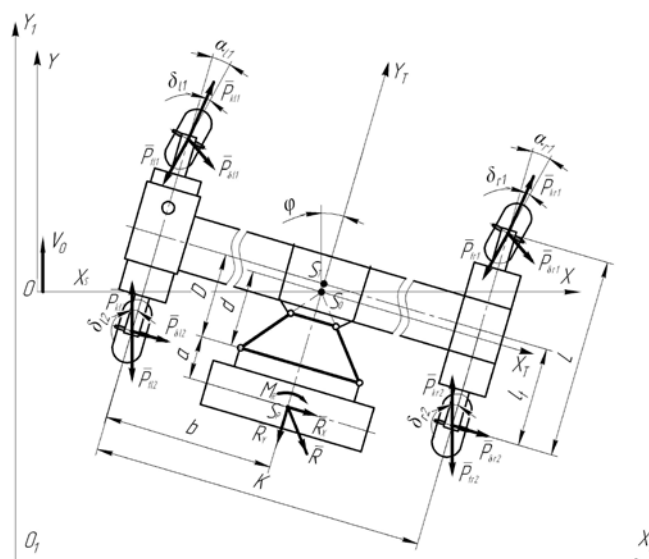


Fig. 2. A scheme of forces acting upon the wide-span vehicle in a horizontal plane with a kinematic method of its control

With respect to the XOY plane the investigated wide-span vehicle with a technological part attached to it has two degrees of

freedom, corresponding to two generalised coordinates: angle φ and abscissa X_S (Fig. 2).

To the number of external forces acting upon the wide-span vehicle in its plane-parallel movement (see Fig. 2) belong:

- the forces of rolling resistance of the frontal (P_{fl1}, P_{fl2}) and the rear (P_{rl2}, P_{rl1}) wheels;
- the tangential traction forces $P_{kl1}, P_{kr1}, P_{kl2}, P_{kr2}$, on the frontal and rear wheels;
- lateral forces $P_{\delta l1}, P_{\delta l2}, P_{\delta r1}, P_{\delta r2}$, which lead to the appearance of displacement angles of the frontal δ_{l1}, δ_{r1} and the rear δ_{l2}, δ_{r2} wheels;
- the main vector R (components R_x and R_y , respectively) and the main moment M_R of forces brought to the centre of resistance (point S_0), acting from the side of the agricultural implements.

The value of force R_x and moment M_R will depend on the turning angle β of the lower links of the wide-span vehicle (Fig. 3).

Taking into account the small value of these angles in Fig. 3 we have:

$$\begin{aligned} R_x &= R[\varepsilon \pm (1-c/l)\beta], \\ M_R &= R(d_0 + D - d)[\varepsilon \pm (1-c/l)\beta], \end{aligned} \quad (1)$$

where ε – an angle between the direction of the traction force of the tractor and the longitudinal axis of symmetry of its technological part;

β – the turning angle of the lower links of the mounted wide-span vehicle;

d_0 – the distance in a longitudinal direction from the point of application of the hook effort to the instantaneous turning centre of the mounted tractor attachment (Fig. 3);

c, l – the design parameters of the mounted wide-span vehicle (Fig. 3).

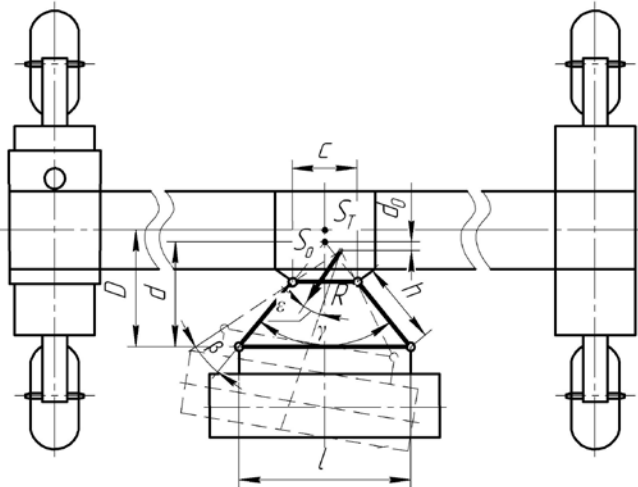


Fig. 3. A scheme of turning of the lower links of the mounted wide-span vehicle attached at angle β

In dependencies (1), the choice of the sign from the proposed record "±" is determined by the correspondence of the direction of angles ε and β . So the "-" sign is accepted when the indicated angles of the agricultural implement change in the process of the work of the wide-span vehicle in antiphase. In case of their in-phase change sign "+" is adopted.

3. Results and discussion

The differential equations of the movement of the wide-span vehicle with respect to plane XOY (see Fig. 2) will look like:

$$\begin{aligned} A_{11}\ddot{X}_S + A_{12}\dot{X}_S + A_{13}\dot{\varphi} + A_{14}\varphi &= f_{11}\alpha + f_{12}\beta + f_{13}; \\ A_{21}\ddot{\varphi} + A_{22}\dot{\varphi} + A_{23}\varphi + A_{24}\dot{X}_S &= f_{21}\alpha + f_{22}\beta + f_{23}, \end{aligned} \quad (2)$$

where $A_{11} = M_T + M_R$;

$$A_{12} = \left(k_{l1} + k_{l2} + k_{r1} + k_{r2} - P_{kl1} + P_{fl1} - P_{kr1} + P_{fr1} - P_{kl2} + P_{fl2} - P_{kr2} + P_{fr2} \right) / V_0;$$

$$A_{13} = \left[(k_{l1} + k_{r1} - P_{kl1} + P_{fl1} - P_{kr1} + P_{fr1})(L - l_t + D - d) + (P_{kl2} - P_{fl2} + P_{kr2} - P_{fr2} - k_{l2} - k_{r2})(l_t - D + d) \right] / V_0;$$

$$A_{14} = -A_{12}V_0; \quad A_{21} = J_{SO};$$

$$A_{22} = \left[(k_{l1} + k_{r1} - P_{kl1} + P_{fl1} - P_{kr1} + P_{fr1})(L - l_t + D - d)^2 + (P_{kl2} - P_{fl2} + P_{kr2} - P_{fr2} - k_{l2} - k_{r2})(l_t - D + d)^2 \right] / V_0;$$

$$A_{23} = -A_{13}V_0; \quad A_{24} = A_{13};$$

$$f_{11} = k_{l1} + k_{r1}; \quad f_{12} = R(1 - c/l); \quad f_{13} = R\varepsilon;$$

$$f_{21} = (k_{l1} + k_{r1} + P_{kl1} - P_{fl1} + P_{kr1} - P_{fr1})(L - l_t + D - d);$$

$$f_{22} = R(1 - c/l)(d_0 + D - a - 2d) + (P_{kl1} - P_{fl1} + P_{kl2} - P_{fl2})b - (P_{kr1} - P_{fr1} + P_{kr2} - P_{fr2})(K - b);$$

$$f_{23} = R(d_0 + D - a - 2d)\varepsilon.$$

In the mathematical model (2) the following designations are accepted: M_T, M_P – masses of the tractor and its technological part, respectively; V_0 – the forward speed of the tractor; J_{SO} – the moment of inertia of the tractor with the technological part relative to point S_0 ; k_{l1}, k_{r1} and k_{l2}, k_{r2} – coefficients of the drag resistance of the frontal and rear wheels of the wide-span vehicle, respectively L, l_t, K, a, b, d, D – the design parameters of the wide-span vehicle the nature of which is clear from Fig. 2 and 3; α_{l1}, α_{r1} – the turning angles of the driven wheels of the tractor at its kinematic method of control.

Applying transformation $d/dt = s$, we will obtain a mathematical model of the plane-parallel movement of the wide-span vehicle in a horizontal plane, recorded in an operator form:

$$\begin{cases} K_{11}X_S(s) + K_{12}\varphi(s) = F_{11}\alpha(s) + F_{12}\beta(s) + F_{13}(s); \\ K_{21}X_S(s) + K_{22}\varphi(s) = F_{21}\alpha(s) + F_{22}\beta(s) + F_{23}(s), \end{cases} \quad (3)$$

where $K_{11} = A_{11}s^2 + A_{12}s$; $K_{12} = A_{13}s + A_{14}$; $K_{21} = A_{24}s$; $K_{22} = A_{21}s^2 + A_{22}s + A_{23}$; $F_{11} = f_{11}$; $F_{12} = f_{12}$; $F_{13} = f_{13}$; $F_{21} = f_{21}$; $F_{22} = f_{22}$; $F_{23} = f_{23}$;

$\frac{d}{dt} = s$ – the operator of differentiation.

The nature of the response of the dynamic system to the control and disturbing impacts is determined by the design and technological parameters of the wide-span vehicle, which enter into the coefficients of the right-hand parts of the system (3). In our case the point of interest is the nature of the response of the control and disturbing impacts of the considered dynamic system at various ratios of the design parameters d and D (see Fig. 3). These parameters determine the distance from the axis that passes through the attachment points of the lower links of the mounted device to the instantaneous turning centre (point S_0) and the centre of the mass (point S_1) of the tractor. Therefore, in the process of mathematical simulation, this ratio of parameters was considered for three cases: $d < D$; $d = D$ and $d > D$.

The output values of the mathematical model (3) are transverse displacements X_S of the wide-span vehicle, and its relative bearing φ .

The controllability of the movement of the wide-span vehicle was estimated by analysing the corresponding transfer functions with the help of the constructed amplitude and phase frequency characteristics.

The transfer function $W^\alpha(s)$, which describes variations in

the relative bearing φ of the wide-span vehicle when the turning angle α of its wheels change, can be presented according to the system of equations (3) as:

$$W^\alpha(s) = \frac{C_{1\alpha}s + C_{0\alpha}}{s(B_2s^2 + B_1s + B_0)}, \quad (4)$$

where $C_{1\alpha} = A_{11}f_{21}$; $C_{0\alpha} = A_{12}f_{21} - A_{24}f_{11}$; $B_2 = A_{11}A_{21}$; $B_1 = A_{12}A_{21} + A_{11}A_{22}$; $B_0 = A_{12}A_{22} + A_{11}A_{23} - A_{13}A_{24}$.

Analysis of the conducted theoretical studies showed that the controllability of the wide-span vehicle of the TGATU design at the speed of its movement $2 \text{ m} \cdot \text{s}^{-1}$ depends little on the change in the distance from the axis that passes through the attachment points of the lower links of the mounted device to the instantaneous turning centre (point S_0) with respect to the centre of mass (point S_T). At low fluctuation frequencies (up to $0.8 \dots 0.9 \text{ s}^{-1}$) of the turning angle α of the frontal wheels of the tractor, in possible variants of the ratio of parameters d and D (Figure 2), we have significant overregulation but at high frequencies - underregulation of the control impact by the dynamic system. Besides the desired fluctuation frequency of the turning angle α of the frontal wheels of the tractor is on the level $0.8 \dots 0.9 \text{ s}^{-1}$, where the value of the amplification coefficient of the input control impact is close to one.

The worst amplitude and phase frequency characteristics of the control impact developed by the dynamic system are observed when $d > D$ since in the operating range of frequencies these characteristics are most distant from the desired ones.

The most desirable amplitude and phase frequency characteristics of the control impact developed by the dynamic system are observed when $d < D$. In this case, in the operating range of frequencies, these characteristics are closest to the desired ones.

On the whole, the following trend emerges: in order to ensure satisfactory controllability of the wide-span vehicle of a new design, the fluctuation frequency of the turning angle of its wheels at a speed of $2 \text{ m} \cdot \text{s}^{-1}$ should be equal to $0.8 \dots 0.9 \text{ s}^{-1}$. Otherwise the amplitude and phase frequency characteristics of the control impact developed by the dynamic system are approximated to the desired ones in the case when the instantaneous turning centre of the wide-span vehicle is in the centre of its mass.

Delayed response of the wide-span vehicle to the control action is the less, the less is distance d . It is another matter that the difference between the obtained phase-frequency characteristics is not significant and generally manifests itself only at the frequencies of the control impact, greater than $0.8 \dots 0.9 \text{ s}^{-1}$.

The transfer function $W^\beta(s)$, which reflects a measurement of the change in the relative bearing of the wide-span vehicle from value β , according to the system of equations (3), can be presented as:

$$W^\beta(s) = \frac{C_{1\beta}s + C_{0\beta}}{s(B_2s^2 + B_1s + B_0)}, \quad (5)$$

where $C_{1\beta} = A_{11}f_{22}$; $C_{0\beta} = A_{12}f_{22} - A_{24}f_{12}$.

From the analysis of the development of a disturbing impact by the dynamic system it follows that with an increase in parameter d the amplitude-frequency characteristics undesirably increase. It is true that this process is more or less significant only at low fluctuation frequencies of angle β ($0.5 \dots 1.3 \text{ s}^{-1}$). The most desirable variant of the ratio of the considered parameters is when $d > D$. The latter condition indicates that the instantaneous turning centre of the wide-span vehicle should be in the centre of its mass.

This condition can be ensured by choosing such a convergence angle γ of the lower links of the mounted wide-span vehicle in which the instantaneous turning centre of the first link will be located in the zone of the centre of mass of the second link. In practice this can be achieved by setting the required value of c (Fig. 3). When $d = D$, parameter c in Fig. 3 should be equal to:

$$c = 1 \left(1 - \frac{2h}{\sqrt{1^2 + 4D^2}} \right). \quad (6)$$

And the convergence angle γ of the lower links of the mounted wide-span vehicle is equal to:

$$\gamma = 2\text{ATAN}\left(\frac{1}{2D}\right). \quad (7)$$

Analysis of dependencies (6) and (7) for the mounted devices of high categories 4N and 4, according to ISO 730: 2009, showed that, increasing distance D from the axis that passes through the attachment points of the lower links to its centre of mass, the required increase in length c varies nonproportionally (Fig. 4). As nonproportional is also reduction of the convergence angle γ of the lower links. On the whole, increasing length D from 0.6 to 2.6 m, c increases from 0.02 m, which practically corresponds to full reduction of the lower links of the wide-span vehicle to one point, up to 0.77 m. In this case angle γ decreases from 85 to 24 deg.

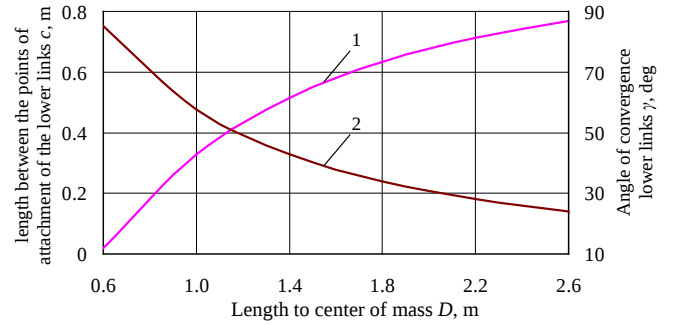


Fig. 4. Dependence of the distance between the points of attachment of the lower links of the mounted wide-span vehicle (1) and the convergence angle of its lower links (2) upon the distance to its centre of mass

The conducted investigations showed that, in order to ensure sufficient angular mobility of the technological part of the wide-span vehicle in relation to its framework, the required distance between the attachment points of the lower links of the mounted device is substantially determined by the distance to its centre of mass. Besides, the proximity of the location of the mounted device to the centre of the mass of the wide-span vehicle is determined by the fact that the required distance between the attachment points of the lower links of the mounted device should be so small that it corresponds to their complete reduction to one point. Therefore, from a position of designing a mounted multi-purpose device for the wide-span tractors (vehicles), one should consider a possibility to configure it, both according to a three-point and a two-point scheme.

4. Conclusion

1. It has been established during the research that the most desirable is such a variant of configuration of the mounted device of the wide-span tractor (vehicle) in which the instantaneous turning centre, formed by convergent rays of its lower links, is in the centre of the mass of the tractor.

2. To ensure sufficient angular mobility of the technological part of the wide-span tractor (vehicle) relative to its framework, the required distance between the attachment points and the convergence angle of the lower links of the mounted device is substantially determined by the distance to its centre of mass. When the latter is reduced, the required distance between the attachment points of the lower links of the mounted device can be so small that it corresponds to their complete reduction to one point.

3. From the position of designing a multi-purpose mounted device for wide-span tractors (vehicles), one should consider a possibility to configure it, both according to a three-point and a two-point scheme.

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EXPERIMENTAL STUDY OF OPERATION INDICATORS OF BEET TOPS HARVESTING MACHINE FOR SUGAR BEET TOPS CONTINUOUS CUTTING

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Abstract: Field experimental studies of a new beet tops harvesting machine, which is front mounted on a wheeled tractor aggregating tractor, are carried out, carrying out a continuous non-sensing cut of the green mass of sugar beet tops. At the same time, a new laboratory-field installation was developed, which consists of a rotor type front mounted rotary type machine with a horizontal axis of rotation, a transverse screw for collecting the cut leaves from all width, and a loading mechanism in the form of a paddlewheel and a loading nozzle. During the field experiments, the tractor moving with a specified speed regime fixed by track measuring wheel, the total cutting height was set in a predetermined interval by means of two pneumatic sensing wheels with adjustment mechanisms. The rotational speed of the rotary cutter, with arched knives mounted on the drive horizontal drum, was varied with the help of replacement asterisks. In the prepared area of sugar beet field, the beet tops was cut at various parameters of the harvesting process, which was then estimated by the size of the remaining tops remnants left on root crops heads and in the aisles. According to the developed program and methodology, statistical processing of the results of experimental studies with the help of a personal computer was subsequently carried out, which allowed obtaining graphical dependencies of the indicated tops residues on the root crops heads, which made it possible to establish performance of new harvesting machine for continuous cutting of sugar beet tops. The established dependencies made it possible to choose rational constructive and technological parameters for harvesting tops by front mounted harvesting machine.

KEY WORDS: BEET TOPS, CLEANING, CONTINUOUS CUT, EXPERIMENTAL STUDIES.

1. Introduction

High-performance and high-quality harvesting of sugar beet tops remains a rather complex and urgent task for sugar beet growing. Recently, the most common in the world is the three-phase method of tops harvesting, which is based on a continuous main cutting of entire tops mass (at width of harvesting machine), its collection and transfer to a vehicle that moves side by side, and then using an individual copy of each head of root crop in a row, after-treatment or after-cutting (or simultaneously: and additional cleaning and pre-cutting with different working organs) heads of root crops from the tops remains. Since these operations are carried out sequentially for root crops of sugar beet in the soil (i.e., at the root) and the harvesting of the tops precedes the operation of digging out the beet roots from the soil, the tops harvesting machines, as independent agricultural machines, or tops harvesting modules, as integral units of sugar beet harvesters are located in the frontal position with respect to the power means (to the tractor – if it is a hinged tops harvesting machine or to the front part of the self-propelled frame beet root harvesters). However, our experimental studies have established that in the process of work, the front-mounted machine placed on the tractor carries out movements in space that are determined by the relief of the surface of the field, the translational speed of the tractor, placement of sensing wheels, relative to the suspension system of the machine, etc., affects the quality of this process. The use of pneumatic wheels as sensing causes the oscillating of cutting mechanism in the vertical plane, which significantly affects the quality of the process – uniform cutting of a tops of vegetable from heads of root crops on all width of cut, its fullest collecting and transportation without losses. The practical solution of this problem determines the relevance of this work.

2. Preconditions and means for resolving the problem

In spite of the large spread of front-mounted beet tops harvest modules of western production of beet harvesters, as well as some designs of domestic front-mounted tops harvesting machines, this almost did not lead to analytical and experimental studies of their

oscillatory motion. However, in part, the task of studying the effect of constructive parameters on the movement along rows of sugar beet roots and the unevenness of the soil surface was tried in the following works [1-6].

2.1. Purpose of the study

The aim of the study is to experimentally determine the rational parameters of cutting the beet tops by developed unit to ensure the required quality of tops harvesting.

2.2. Methods of research

Experimental studies were conducted on the fields of the research farm "Olenovskoe" in the Fastovsky district of Kiev region. The object of experimental studies was the working process of tops cutting by developed unit. The conditions for carrying out the studies, which were determined according to known methods [7-10], are given in the table.

To implement the program of experimental research of the technological process of tops removal with the use of a tops harvesting machine, a laboratory-field experimental installation was developed (Fig. 1), which is front mounted on a tractor-type arable and row-crop wheeled tractor of drawbar category 3.0, and the cutter apparatus is made in the form of a horizontal rotor [5]. The machine makes it possible to realize a continuous, non-sensing cut of the main mass of beet tops with its further loading into the vehicle.

Laboratory-field experimental installation (Fig. 1) consists of a wheeled arable tractor I and front mounted toppler II, which contains a frame 1, an attachment 2, a copying wheel 3, a rotary cutter apparatus 4, a transport device 5, a loading device 6, as well as the drive 7.

The laboratory-field experimental setup [5] allows to carry out experimental researches of the given tops harvesting machine in full accordance with the adopted program and methodology (Fig. 1), with the possibility of changing the factors within the established boundaries: rotor speed with the drive mechanism and tachometer control; speed of movement of the harvester with the help of the tractor's gearbox and control of its actual value by track measurement wheel; The height of the rotor installation with a lever mechanism with the control by means of a ruler.

Table 1. Conditions for carrying out laboratory-field experimental research of tops harvesting machine

Index	Value
Soil type	Black soil with middle organic matter (OM), middle clay
Soil hardness in the rooting area, MPa	0.94...2.3
Soil moisture in the area of root crops, %	18.0... 21.6
Average beet root yield, t·ha ⁻¹	46.4
Average yield of the tops, t·ha ⁻¹	44.8
Average density of plantings of sugar beets, pcs· ha ⁻¹	112000
Maximum deviation of position of sugar beet roots from the conventional line, mm	30...40
Position of sugar beet root heads above the soil surface, mm	0...70
The form of bundles of tops on the heads of root crops: – cones – a half-socket	there is a significant amount of dry and fallen stalks of the foliage) more than 90% 10%
Soil relief	flat
Maximum gradients of the relief, deg	Not more than 2...4

Laboratory-field experimental installation (Fig. 1) consists of a wheeled arable tractor I and front mounted topper II, which contains a frame 1, an attachment 2, a copying wheel 3, a rotary cutter apparatus 4, a transport device 5, a loading device 6, as well as the drive 7.

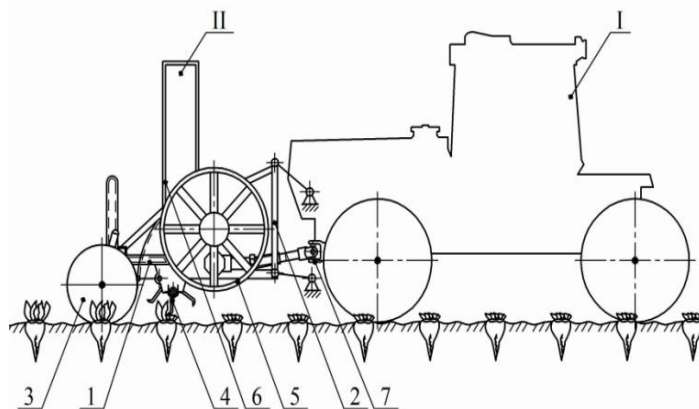


Fig. 1. – The design diagram of the laboratory-field experimental installation:

I – wheeled arable and row-crop tractor; II – front-mounted beet tops harvesting machine; 1 – frame; 2 – the hinged device; 3 – a sensing wheel; 4 – rotary cutter apparatus; 5 – transport device; 6 – the loading device; 7 – drive

The laboratory-field experimental setup [5] allows to carry out experimental researches of the given tops harvesting machine in full accordance with the adopted program and methodology (Fig. 1), with the possibility of changing the factors within the established boundaries: rotor speed with the drive mechanism and tachometer control; speed of movement of the harvester with the help of the tractor's gearbox and control of its actual value by track measurement wheel; The height of the rotor installation with a lever mechanism with the control by means of a ruler.

Based on the calculations performed, previous studies and analysis of a priori information, the levels of variation of factors were established:

- rotational speed of the machine rotor: 500, 750, 1000 rpm;
- speed of the movement of the harvester: 0.5, 1.5, 2.5 m·s⁻¹;
- the height of the cut of the foliage: 0.02, 0.06, 0.10, 0.15 m.

The quality of the work, as indicated above, was taken by tops remnants on the root crops heads, in g·m⁻², which were determined by harvesting all residues (including those not cut off from the heads of the root parts of a part of the foliage) from the area of 1·m² after pass of experimental installation (Fig. 2) and weighing on electronic scales with an accuracy of 1.0 g.



Fig. 2. – Determination of the quality of the removal of the tops by research beet tops harvesting machine

The power tool (tractor), according to its technical characteristics, must provide operation of tops harvesting machine at the required working speeds and adjust to the corresponding track width [12].

The results of the experimental studies were processed according to the known method of statistical processing of the research data [7, 9, 11] with further presentation in the form of functional and graphical dependencies, and also with the application of applications for the PC.

3. Results and discussion

As a result of the experimental studies carried out according to the adopted methodology, the following dependencies were obtained.

For a more complete description of the process of tops removing by rotary harvesting machine, due to the processing of the results of a multifactor experiment, a mathematical model is obtained in the form of regression equation of second degree:

– in natural form:

$$Q = -177,593 - 0,24224n + 530,8054V + 8680,805h + 0,000179nn - 109,767VV - 6795,18hh - 0,09602n - 1159,51Vh - 4,22748nh + 2,158437nVh, \quad (1)$$

– in coded form:

$$Y = 587,2724 - 50,5026X1 + 168,38335327X2 + 327,8374X3 + 11,18333X1X1 - 28,7833X3X3 + 21,86292X1X2 + 29,85444X2X3 - 16,0845X1X3 + 35,07582X1X2X3. \quad (2)$$

where V – is the forward speed of the tractor, m·s⁻¹; n – is the rotational speed of cutting rotor, rpm; h – cutting height of tops, m.

Graphical interpretation of the indicated regression equations (1) and (2) presented in the form of response surfaces (Fig. 3 and Fig. 4).

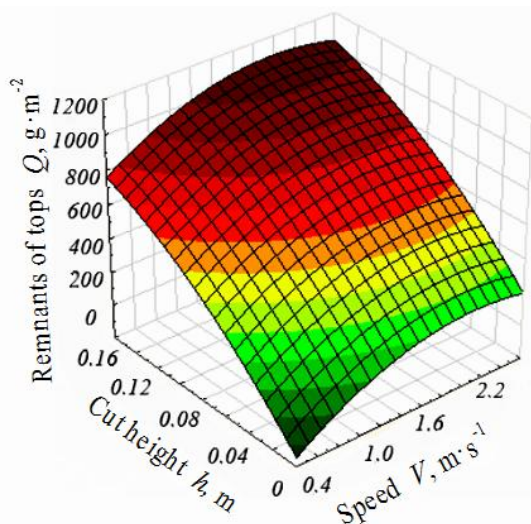


Fig. 3. – The response surface of the dependence of tops remnants on root crop head surface on the speed of movement of tops harvesting machine and the cut height at a rotor speed of 500 rpm

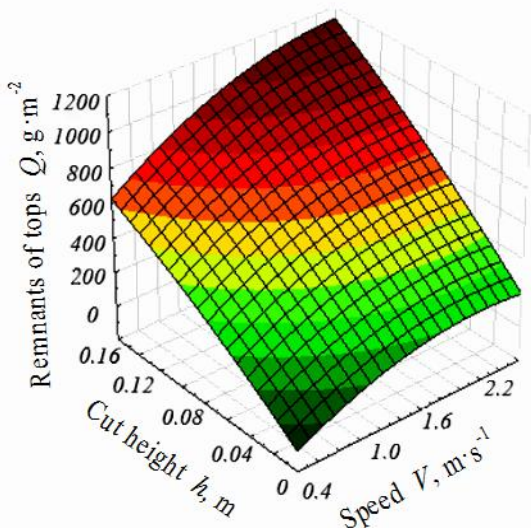


Fig. 4. – The response surface of the dependence of tops remnants on root crop head surface on the speed of movement of tops harvesting machine and the cut height at a rotor speed of 1000 rpm

On the basis of the factor analysis of regression equations (2), rational parameters of the process are determined, at which a qualitative removal of the tops by a rotary cutting apparatus will be achieved at the cutting height is 0.02 m, the speed of the machine is 1.5...2.0 m·s⁻¹, the rotor speed is 1000 rpm.

4. Conclusions

1. On the basis of the analysis of the empirical mathematical model in the form of the regression equation of the haulm removal process, it is established that the height of the cut has the greatest influence on the mass of the tops of the tops on the surface of the root heads of the root crops, while the lowest is the rotor speed of the cutter apparatus. With an increase in the speed of the machine and the height of the cut, the mass of the remains of the tops on the head of the root crops will also increase, while increasing the rotor speed will, on the contrary, reduce the mass of the remains of the tops.

2. It has been established that the rational values of process parameters at which qualitative tops removal will be achieved by a rotary cutter apparatus is the cutoff height is 0.02 m, the speed of the machine is 1.5...2.0 m·s⁻¹, the frequency rotor rotation – 1000 rpm.

3. Based on the results of field experimental studies of the operation of tops harvesting machine, which is front-mounted on the arable and row-crop tractor, when harvesting the tops with a continuous cutting, it can be concluded that machine performance indicators meet the agrotechnical requirements, and therefore the expediency of its application in production conditions is proved.

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VACUUM CONTROL OF MILKING MACHINES BY USING THE FREQUENCY CONVERTER AND THE REDUCING VALVE

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Abstract: The article deals with the milking machine vacuum system which is designed for the vacuum level control with a special reducing valve. Vacuum Milking machines with the vacuum pumps without frequency converter have the constant rotation speed and vacuum level is controlled by the control valve. By using the frequency converter for controlling electric motor of vacuum pump the electric energy consumption significantly decreases but the vacuum fluctuation is higher. The vacuum pump of the designed vacuum system is connected to the air container. The vacuum level of the air container is controlled by the frequency converter on a higher level than the level of the milking vacuum. Newly designed reducing valve is placed between the air container and the vacuum pipeline system of the milking machine. The reducing valve controls the air which flows through it and the vacuum is reduced on the required level and is stabilized. But the energy consumption is a little bit higher.

KEY WORDS: MILKING MACHINES, VACUUM PUMP, VACUUM LEVEL CONTROL, FREQUENCY CONVERTER

1. Introduction

If a constant vacuum is to be obtained by applying a standard regulation method of vacuum magnitude, the control valve must suck so much atmospheric air into the vacuum system that the sum of the air amount sucked in by the milking machine and the control valve within a time interval is constant. The vacuum pump works at a full efficiency irrespective of the actual consumption by the milking machine. For this presently, frequency converters are commonly used to drive the vacuum pumps of milking machines as this result in major electricity consumption reduction. Their frequent using is encouraged by their relatively low price and the resulting short payback periods. Rasmussen states that the payback period is approx. 4.8 years [RASMUSSEN, 2005]. Olejník later gave a payback period of 2 years [OLEJNÍK et al., 2008]. Since then, the prices of frequency converters have dropped even further and their payback period is currently even less than one year. Kudělka features already only about 200 days [KUDĚLKA et al., 2012]. The use of frequency converters results in power consumption reduction ranging around 50 %. Olejník gives a cost reduction value of 50 - 58 % [OLEJNÍK et al., 2008], Kudělka features 57 % [KUDĚLKA 2016]. To a certain extent, the reduction percentage is affected by the vacuum pump delivery given the size of the milking plant. If an unnecessarily high capacity vacuum pump is selected, the reduction appears to be high, too. Rasmussen measured the absolute reduction and states that the automatic milking system DeLaval VMS showed a saving of 20 kWh of electric power over 24 hrs of frequency converter-driven vacuum pumps being in use [RASMUSSEN, 2005]. The vacuum pump revolution control has been dealt with for a relatively long time [DUNN, 1996] and research in the field of milking has focused on vacuum control in the individual teat cups. [STRÖBEL et al., 2012, STRÖBEL et al., 2013] In practice, certain problems are still encountered when using the frequency converters.

2. Prerequisites and means for solving the problem

Vacuum control by changing the revolutions entails certain deterioration in the vacuum stability. Vacuum fluctuation is reflected in the deformation curve of the teat liner. The teat liner deformation curve in dependence on vacuum was described by Karas [KARAS et al., 2003]. Greater vacuum fluctuation is determined by the fact that air flow rate changes, i.e. during the teat cup application, occur very fast and changes in the electric motor and vacuum pump revolutions take place at a slower pace, which is

caused by the rotating part inertia. Therefore, milking machines with frequency converters are also fitted with a conventional control valve, which is set to a value that is by ca. 1 kPa higher than the vacuum level set for the frequency converter control. If the air flow rapidly drops, the vacuum would be increasing for a short period of time. The control valve prevents from the increase exceeding 1 kPa. Contrariwise, when the flow rate increases, no system preventing from the vacuum drop is available and the vacuum changes depend on the time over which the vacuum pump revolutions reach the desired value.

To validate the proper functioning of the stabilisation device, vacuum fluctuation was measured in a tandem milking parlour 2x3 with very good results [FRYČ et al., 2016]. The same device was tested in large herringbone milking parlour 2x13, but results was not satisfactory [FRYČ et al., 2015]. Therefore, was made other newly designed reducing valve. The objective of this study is to validate the applicability of the manufactured stabilisation device prototype in laboratory.

3. Solution of the examined problem

One of the basic demands for milking machines is to maintain a constant vacuum level if air consumption by milking machine is changed. The reduce valve prototype was manufactured by Utility model 14020 [FRYČ, 2004], but was modified for higher air flow rate. To validate the proper functioning of the stabilisation device with reduce valve, vacuum fluctuation was measured in laboratory, where was installed milking machine by Fig. 1.

The vacuum pump of the designed vacuum system is connected to the air container AC1. The vacuum level of the air container AC1 is controlled by the frequency converter on a higher level than the level of the milking vacuum. Newly designed reducing valve is placed between the air container AC1 and the vacuum pipeline system of the milking machine (air container AC2 at experimental device). The reducing valve controls the air which flows through it and the vacuum is reduced on the required level and is stabilized. The experiment milking plant is equipped with oil lubricated rotary vane vacuum pump with an actual delivery of 1564 l.min^{-1} at a 50 kPa vacuum. Working vacuum in experiments was 45 kPa.

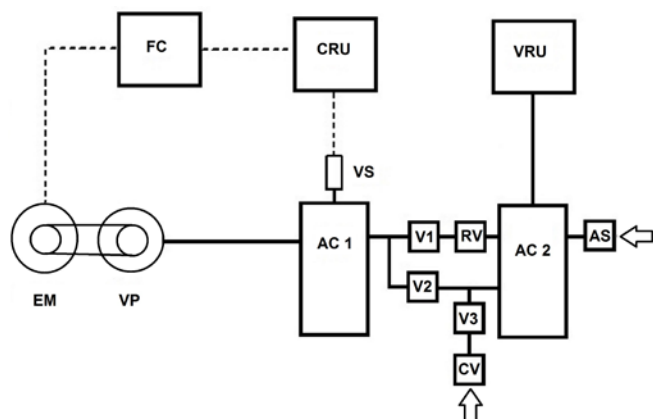


Fig. 1 Scheme of experimental device

FC – frequency converter, CRU – control and record unit, VRU – vacuum record unit, EM – electric motor, VP – vacuum pump, VS – vacuum sensor, AC1, AC2 air container, V1, V2, V3 – closing valve, RV – reducing valve, CV – control valve, AS – air sucking control device

The milking machine vacuum system capacity (pipe lines, air containers) totals 254 l. Three vacuum control alternatives were measured:

1. Control valve
(valve V1 closed, valve V2 open, valve V3 open)
2. Frequency converter
(valve V1 closed, valve V2 open, valve V3 closed)
3. Frequency converter complete with the stabilisation device (reducing valve prototype)
(valve V1 open, valve V2 closed, valve V3 closed)

The vacuum fluctuation measurement was conducted by means of the Pulsatortester PT IV instrument measuring the vacuum for a period of 60 s and over this time interval we determined the maximum, average and minimum vacuum values. The vacuum fluctuation was then calculated as a difference between the maximum and minimum value.

To conduct the measurements in objectively identical conditions, measurement was carried out with the air consumption by a precisely defined variable flow rate (Q_1 , Q_2 , Q_3) see Fig.2. An electromagnetic valve connected instead of the collector to the central suction piece provided for the variable flow. The operation of the electromagnetic valve was controlled by a time relay. 15 s opening intervals alternated with 15 s closing intervals. Once in the open position, the valve sucked 100 l min^{-1} . The total air flow was then gradually increased by sucking in air via three different calibrated openings. More air got into the milking machine vacuum system through the leaking spots. The calibrated openings sucked in 400 l min^{-1} , 600 l min^{-1} and 800 l min^{-1} . The measurement determined that leakage sucked in 22 l min^{-1} .

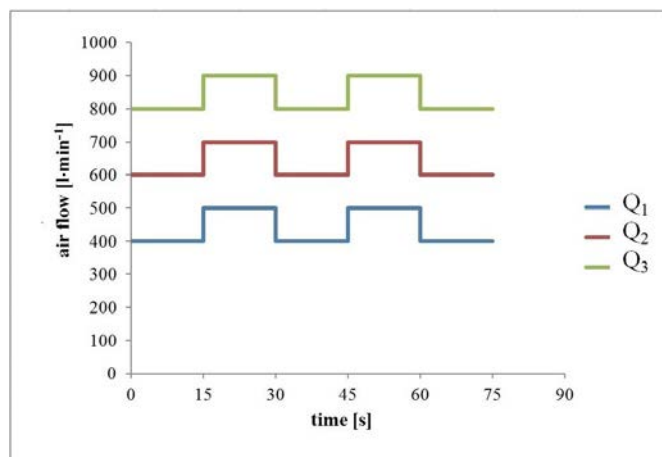


Fig. 2 Variable flow rate of air sucked in by the milking machine

Pressure difference is necessary for reduce valve using. The vacuum level of the air container AC1 is controlled by the frequency converter on a higher level than the level of the working vacuum. Three different levels of vacuum were tested 48 kPa, 51 kPa and 54 kPa. Electric energy consumption was recorded during all experiments.

Each measurement was repeated ten times and the measured values were statistically processed. To process the data, a use was made of the dispersion analysis which was employed to determine the statistical significance of the difference between the measured values determined by the individual vacuum control methods. The method of subsequent testing using the Tukey range test assessed the statistical significance of differences between all the measurement combinations.

4. Results and discussion

When the vacuum fluctuation measurement was carried out under a precisely defined variable air flow rate, the lowest values were achieved when the control valve was used. The average difference between the maximum and minimum vacuum value was 1.1 kPa.

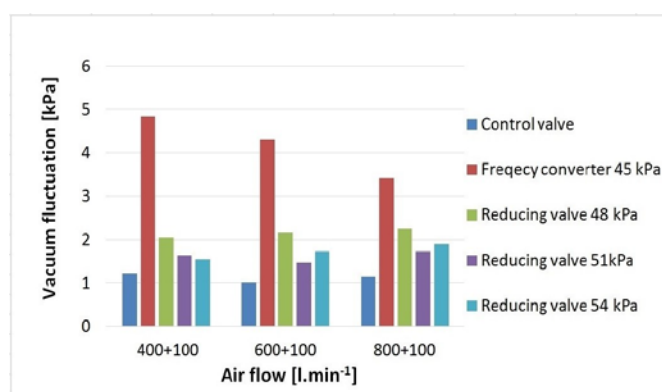


Fig. 3 Vacuum fluctuation at measurement

During frequency converter control the first measurement (flow rate Q_1) was fluctuation 4,8 kPa. With the increasing flow rate, it decreased to values 4.3 kPa (flow rate Q_2) and 3.4 kPa (flow rate Q_3). By connecting the prototype of the stabilisation device with reducing valve was indicated that the vacuum fluctuation is independent on air flow rate, but depends on the vacuum level at air container AC 1. When the vacuum level in the air container AC 1 is

48 kPa the average fluctuation is 2.1 kPa. When the vacuum level in the air container AC 1 is 51 kPa the average fluctuation is 1.6 kPa. When the vacuum level in the air container AC 1 is 54 kPa the average fluctuation is 1.7 kPa. Olejník and Pavelková and other authors state that the vacuum fluctuation during the frequency converter control goes up to 2 kPa when being used on farms [OLEJNÍK et al., 2008]. The comparison of our results with the experiments on farms isn't appropriate. The resembling measurements like ours were carried out by Kudělka and his results are similar [KUDĚLKA 2016].

To process the data, we employed the dispersion analysis based on which we determined that differences in the measured values between the individual vacuum control methods are statistically significant. We obtain following results: We determined the statistically highly significant difference in the measured values between the individual vacuum control methods. Difference between individual measurements with reducing valve are statistically significant only between vacuum levels 48 kPa and 51 kPa. Between other vacuum levels there are not statistically significant differences.

Comparison of electric energy consumption was carried out between vacuum control by frequency converter and vacuum control by reducing valve variations. The results of the electric energy consumption measurements are shown in Fig. 4. When the vacuum level increases the electric energy consumption increases as well. With the increasing flow rate, the differences between values of electric energy consumption increase. The use of frequency converters on farms results in power consumption reduction 50 % or more. [OLEJNÍK et al., 2008], [KUDĚLKA 2016]. We calculated the expected saving of the energy consumption. We supposed that the energy saving is 50 % in case the frequency converter is used for a vacuum control. The calculated energy saving is 46.5 % when vacuum level at air container AC1 is 48 kPa. The calculated energy saving is 42.7 % when the vacuum level at air container AC1 is 51 kPa. The calculated energy saving is 38.2 % when vacuum level at air container AC1 is 54 kPa.

To process the data, we employed the dispersion analysis based on which we determined the statistically significant difference in the measured values between the individual vacuum control methods. After that we tested the differences between the specific control methods at the identical flow rate. The testing method – Tukey test – revealed statistically significant differences between all measurements combinations at all the set flow rates.

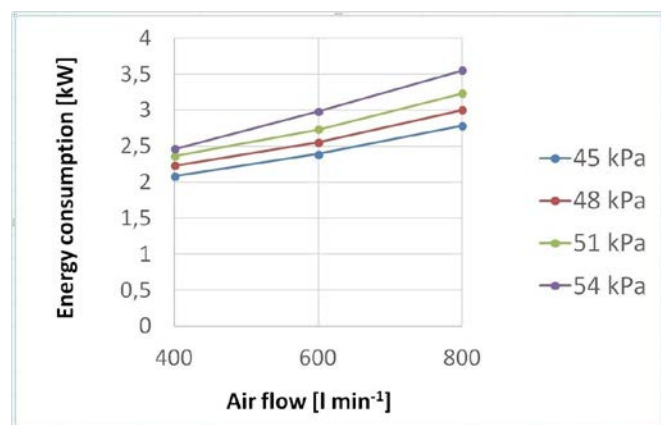


Fig. 4 Energy consumption of vacuum pump electric motor

5. Conclusion

The measurement we performed confirms that if a frequency converter is used, the vacuum fluctuation is at higher level as in case a control valve is used. Likewise it has been demonstrated that the proposed prototype of the stabilisation device can reduce the fluctuation. The achieved results are not as good as in case the control valve is used. On the other hand the vacuum fluctuation reduction nearly by 50 % is satisfactory. Higher energy consumption is the disadvantage of the vacuum control device with reducing valve. Saving of the energy consumption decreases from 50 % to 46.5 % in the best case. We have to choose a convenient variant with optimum vacuum fluctuation and energy consumption saving.

Acknowledgment:

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INVESTIGATION OF THE GRAIN SEPARATION PROCESS IN THE THREE-DRUM THRESHING-SEPARATING DEVICE OF A COMBINE HARVESTER

ИССЛЕДОВАНИЕ ПРОЦЕССА СЕПАРАЦИИ ЗЕРНА В МОЛОТИЛЬНО-СЕПАРИРУЮЩЕМ УСТРОЙСТВЕ ТРИБАРАБАННОГО ТИПА ЗЕРНОУБОРОЧНОГО КОМБАЙНА

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Abstract: The purpose is improving the efficiency of the technological process of threshing and separating the grain pile in a multi-drum threshing-separating device due to the substantiation of the technological modes of the work of threshing drums. The influence of technological parameters of the three-drum threshing-separating device of a grain-harvesting combine on the process of grain separation has been studied experimentally. Models of the grain separation process are obtained depending on the speed of rotation of the drums and the supply of technological material for threshing, based on the results of the experimental research. modeling the process of grain separation depending on the speed of rotation of the drums and the supply of technological material for threshing allows to determine the quantity and quality of the mass that each of the three drums separately threshed.

KEYWORDS: COMBINE HARVESTER, THRESHING-SEPARATING DEVICE, GRAIN SEPARATION PROCESS, GRAIN MATERIAL

1. Introduction

Improvement of the designs of grain harvesters threshing devices is mainly based on the analysis of the results of experimental studies and testing of various variants of existing models [1, 2]. Such a way of creating a threshing and separating system (TSS) of combine takes a long time and requires high labor and resources. The methods of justifying the rational parameters of the working units of the combine threshing device are promising due to the development of mathematical models of threshing and separation processes [3-7].

Currently, there is a tendency to use rotary, multi-drum, hybrid threshing devices in order to increase the throughput of the combine as a whole [10, 11]. The study of the operation conditions and modes of the combine threshing and separation system, which provide increased separating capacity, and simultaneously reduce the load of one of the most vulnerable unit of the combine such as a straw walker [12], seem to be a priority.

The urgency of the research is due to the necessity of developing universal methods for analyzing multi-drum threshing systems of grain-harvesting combines to determine the quantity and quality of the mass that has been threshed and separated at each operation of the technological process. Improvement of the designs of threshing and separation system occurs to reduce the level of grain damage. Under such conditions, the efficiency of grain production systems is substantially increased [8, 9].

2. Preconditions and means for resolving the problem

Known for the designs of grain combines which in the threshing and separation system contain several drums [13, 14]. The use such designs of the combine harvesters makes the increase in the productivity of the threshing device, which is mainly due to the improvement of grain separation processes. Noted designs include various combinations of drums and concaves.

Of the well-known design and technological schemes of grain threshing and separating it is worth noting the designs of the TSS that use a differential threshing method. The peculiarity of this threshing method [15, 16] is that for the threshing and separation of grain the appropriate modes of operation of the threshing drum are chosen. More moderate modes are set for the first drum and more rigid for subsequent drums (rotors). Due to such conditions the amount of injured grain is reduced and its separation is improved.

Widespread use of the differential method of grain threshing and separation indicates the prospect of conducting research in the

direction of optimization and substantiation of the threshing drums operating modes which would allow intensifying the process of grain threshing and separation and reducing its damage and injury.

To describe the threshing process, a number of empirical dependencies were obtained, which used mainly experimental-theoretical research methodologies [5]. The disadvantages of these methods include the multivariance of the proposed solutions, many of which can be used only in a narrow range of variables and for individual elements of the designs of TSS. In addition, in these works insufficient attention is paid to the study of the technological process of threshing and separation of multi-drum TSS, consisting of two or more drums [5].

3. Purpose of the study

The main aim of our study is improving the efficiency of the technological process of grain threshing and separation in a multi-drum threshing and separating device due to the substantiation of technological modes of operation of threshing drums.

4. Results and Discussion

Consider the technological process of grain mass threshing and separation on the example of a threshing device of combine KZS "Slavutich" with a three-drum TSS (Fig. 1).

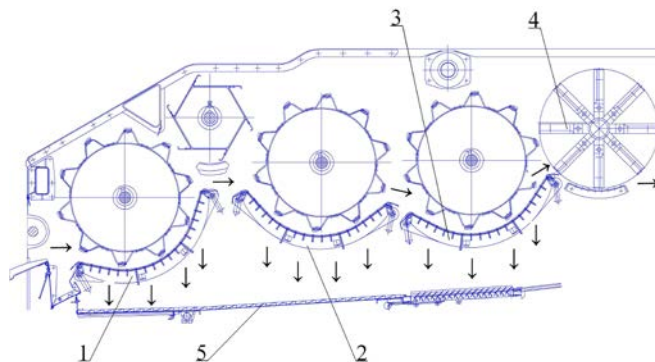


Fig. 1. - Scheme of the threshing device of the combine harvester "Slavutich" with a three-drum TSS: 1, 2, 3 - concaves of the threshing device; 4 - chopper; 5 - shaker pan; 6 - cleaning system.

The structural scheme of a combine harvester KZS "Slavutich" with a three-drum TSS is shown in Fig. 2. According to the technological process, the cut cereal stalks are fed by a header to the slope conveyor and further to the concave of the threshing device 1 (fig. 2). The Q_1 symbol indicates the proportion of grain that is fed to the threshing and separation in the TSS. In the steady-state operating conditions of the combine and the absence in its design of devices for preliminary threshing of grain, $Q_1 = 1$.

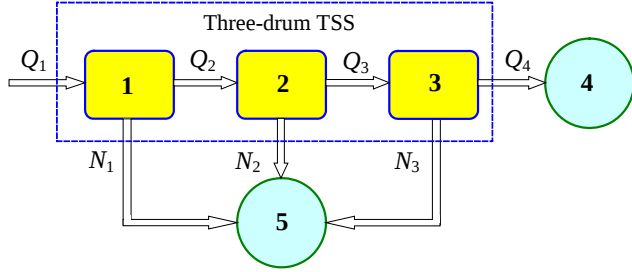


Fig. 2. - The structural scheme of a threshing device of a combine KZS "Slavutich" with three-drum TSS.

The threshing process begins on the first drum. Under such conditions, part of the mass N_1 is sifted through the concave 1, gets to the shaker pan 5 and is directed to the sieves of the cleaning system. A part of the mass Q_2 that has not threshed on the first drum is fed to the second drum, where it further threshing and separation takes place.

On the second drum some of the mass N_2 is sieved through the concave 2 and falls on the shaker pan 5. The mass Q_2 which has not been threshed on the first and second drums is fed to the third drum. At this stage, the mass N_3 is sieved through the concave 3 and also falls on the shaker pan 5.

The mass Q_4 is a mixture of not sifted free grain, unthreshed grain (loss in straw) and non-grain part. This mass is fed to a shredder 4, which evenly spreads it on the field. This mass can be transported to a straw hopper.

The technological regime of the three-drum MSS operation can be expressed conventionally in the following form:

$$\begin{cases} Q_2 = Q_1 - N_1 \\ Q_3 = Q_1 - N_1 - N_2 \\ Q_4 = Q_1 - N_1 - N_2 - N_3 \end{cases} \quad (1)$$

where Q_1 , Q_2 , Q_3 and Q_4 are the proportion of the grain coming with the mass into the first, second, third drum and chopper (relative loss of grain in straw) of three-drum TSS respectively;

N_1 , N_2 and N_3 are proportion of the grain separated by the first, second and third threshing drum, respectively.

According to the results of experimental studies the dependences of the influence of the angular velocity of the rotation of the drums and the reduced mass feed on the grain separation value along the length of the concave of the three-drum TSS are established. The research was carried out on the fields of the experimental farm on the harvesting of winter wheat by direct harvesting using a laboratory-field installation mounted on an experimental grain harvester KZS "Slavutich" with a three-drum TSS (Fig. 1).

The experiments were conducted according to a matrix plan for three factor experiments. Variable factors affecting the performance of a three-drum TSS were: angular velocity of threshing drums ω , rad/s; load of combine harvester q , kg/s; length of the concave area that the mass passes through during the threshing and separation l , m.

According to the results of the experimental studies, regression equations (2-4) were obtained which adequately describe the influence of the investigated factors on the performance indicators of a three-drum TSS:

$$Y_1 = 1 - e^{(0,022353 \cdot \omega_1 - 0,069613 \cdot q) \cdot l^{1,524049}}; \quad (2)$$

$$Y_2 = 1 - Q_2 \cdot \left(e^{(0,064209 \cdot \omega_2 - 0,322064 \cdot q) \cdot l^{1,041219}} \right); \quad (3)$$

$$Y_3 = 1 - Q_3 \cdot \left(e^{(0,047084 \cdot \omega_3 - 0,141601 \cdot q) \cdot l^{1,242060}} \right), \quad (4)$$

where Y_1 , Y_2 and Y_3 are particles of grain threshed and separated by the length of the drum concave of the first, second and third drum.

The adequacy of these models is verified using Fisher's criterion. The results of the calculations indicate that the given regression equations are adequate at the confidence level of 0.95.

In fig. 3 shows the response surface, which determines the effect of feeding the grain and straw mass and the angular velocity of rotation of the threshing drum on the proportion of grain that is separated by the first concave of the three-drum TSS.

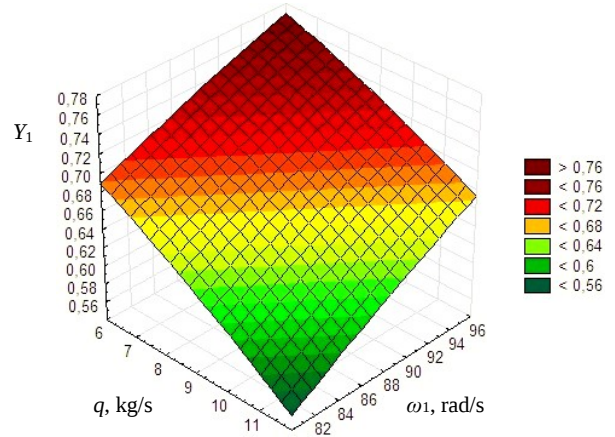


Fig. 3. - The dependence of the grain-straw mass (q) and the angular velocity of the threshing drum rotations (ω_1) on the proportion of the grain that is separated at the length L of the first concave of the three-drum TSS.

On the basis of the analysis of dependencies in Fig. 3 it is established that an increase in the supply of grain and straw mass (q) leads to a decrease in the proportion of separated grain. Thus, when the grain-straw mass was fed ($q = 6$ kg/s), the proportion of the separated grain was 0.69, and when supplied ($q = 12$ kg/s) was 0.55, respectively. The increase in grain-straw supply twice leads to a corresponding decrease in the fraction of the separated grain by almost 20%.

An increase in the angular velocity of the threshing drum rotations (ω_1) leads to an increase in the proportion of grain that is separated at the length L of the first concave of the three-drum TSS (Fig. 3). Thus, at the angular velocity of rotation of the threshing drums ($\omega_1 = 82$ rad/s), the proportion of separated grains was 0.55. The value of the angular velocity of rotation of the threshing drums ($\omega_1 = 96$ rad/s) corresponds to the proportion of the separated grain at the level of 0.69. That is, an increase in the angular velocity of rotation of the threshing drums by 17% leads to a corresponding increase in the proportion of separated grain by 25%.

Simulation of the threshing and separation process of the grain mass of a three-drum TSS was performed in the MathCAD Professional 2000 software environment. Based on the studies carried out models were obtained that made it possible to establish the dependences of the rotation speed of the corresponding drum and the feeding of grain mass on grain separation along the length of concave of each of the three drums.

Analysis of the obtained dependences allowed concluding that the overall level of grain damage in a three-drum TSS can be reduced by reducing the circumferential speed of the first drum by 20-25%. Under such conditions the separating ability of the first

drum decreases by 18% -22% and the corresponding increase in the separating capacity of the second and third drum of the three-drum TSS (Fig. 4).

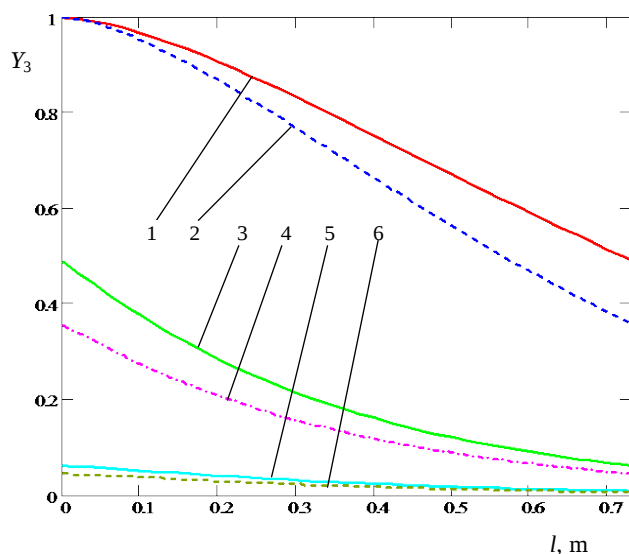


Fig. 4. - Dependence change in the amount of grain remains in the grain and straw mass along the length of the concave of the first (1, 2), second (3, 4) and third (5, 6) threshing drum for the circumferential velocity value of drum $V_B = 24.5$ m/s (1, 3, 5) and $V_B = 31.5$ m/s (2, 4, 6).

The linear speed of rotation of the threshing drums significantly affects the separating ability and the value of grain losses over the TSS of the combine. In Fig. 5 shows the estimated dependences of influence the rotation speed of the second and third drums on the grain losses for the three-drum TSS. The value of the velocity of the second and third drums in the range of 30 - 35 m/s corresponds to the value of grain losses for TSS at the level of 0.5% - 1.1% (Fig. 5). Under the conditions of reducing the velocity of the second and third drums to the values at the level of 28 - 29 m/s there is an increase in grain losses for TSS that exceed the allowable level established by agricultural requirements.

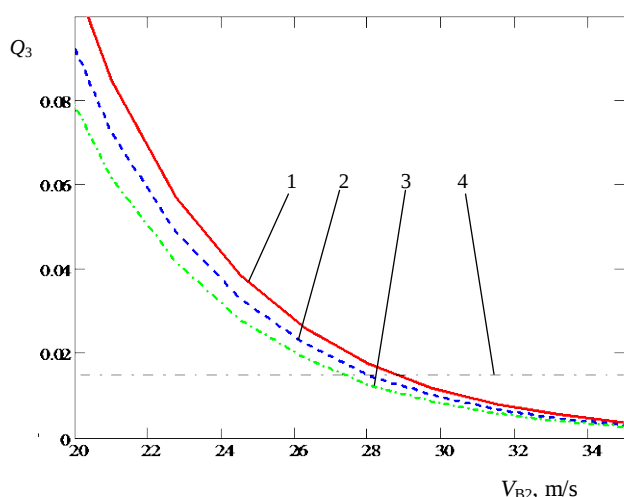


Fig. 5. - Dependences of the influence of the velocity of rotation of the second and third drum of a three-drum TSS on the grain loss: the value of the circumferential velocity of the first drum 1 - $V_{B1} = 24.5$ m/s; 2 - $V_{B1} = 28.0$ m/s; 3 - $V_{B1} = 31.5$ m/s; 4 - the level of grain losses, set by agricultural requirements.

Dependences of the influence of the rotation velocity of the first drum on the level of grain losses for the three-drum TSS for different values of the rotation velocity of the second and third drums are shown in Fig. 6.

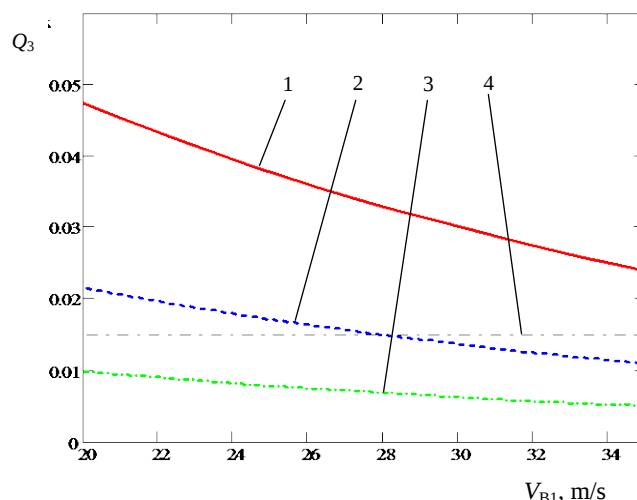


Fig. 6. - Dependences of the influence of the rotation velocity of the first drum of the three-drum TSS on the loss of grain: the circumferential velocity of the second and third drum 1 - $V_{B2} = 25$ m/s; 2 - $V_{B2} = 28$ m/s; 3 - $V_{B2} = 31$ m/s; 4 - the level of grain losses, set by agricultural requirements.

It should be noted that the increase in the circumferential velocity of the first drum leads to a decrease in the grain losses for the TSS of combine harvester (Fig. 6). With the value of the circumferential velocity of the second and third drums $V_{B2} = 31$ m/s, the grain losses for the three-drum TSS do not exceed the permissible level set by agricultural requirements on all ranges is the change in the velocity of the first drum.

4. Conclusions

An experimental calculation method is developed for determining the proportion of threshed and separated grain depending on the concave length of the multi-drum TSS based on the results of simulation the combined process of transportation, threshing and separation of grains due to the interaction of drum with grain-straw mass.

The models of the technological process of grain separation have been developed, which made it possible to establish the dependence of the influence the structural and technological parameters on the grain loss. The dependences of the effect of the rotation velocity of the drums and the reduced mass feeding on the grain separation value along the length of the concaves (first, second and third) of a three-drum TSS are determined. Regressions equations are obtained that establish the proportion of threshed and separated grain depending on the length of the concave of each of the three drums.

It is noted that the overall level of grain damage in a three-drum TSS decreases with the corresponding reduction of the circumferential speed of the first drum by 20 - 25%. Under these conditions, the separation capacity of the first drum is decreases by 18% - 22%, and the separation capacity of the second and third drum of the three-drum TSS is correspondingly increasing.

To ensure the minimum grain losses for the three-drum TSS the value of the circumferential speed for the second and third drums in the conditions of harvesting grain crops should be within 29 - 31 m/s.

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EXPERIMENTAL STUDY OF VACUUM DRYING SEEDS OF GRAIN CROPS

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Abstract: The article is devoted to the problem of drying seeds without their thermal injury by applying vacuum technology. In this paper work presents experimental studies of vacuum drying of seeds of grain crops on the example of maize seeds, during which the safe drying regimes were studied, in which the most intense water separation occurs. It has been found that by heating the batch of seeds to 30° C and subsequent evacuation to a pressure of 4 kPa with bubbling under atmospheric pressure is most intense dehumidification at a rate of 0.259%/min.

KEYWORDS: VACUUM DRYING, RESIDUAL PRESSURE, SEEDS, HEAT, TEMPERATURE OF HEATING SEEDS, DEHUMIDIFICATION, DRYING MODE.

1. Introduction

Modern technologies for drying seeds use heating, as the process of moisture transfer from the inner layers of the seed to the external layers is intensified, as is the evaporation of moisture from the surface. As a rule, the heating of seeds occurs through the action of a temperature field with an amplitude value of about 80 ° C, which results in individual seeds before the biochemical processes that cause a change in their structure and chemical composition. But in the same time, due to the difference in the properties of individual seeds, the thermal diffusivity of the seed layer has a nonlinear character, which contributes to the uneven drying and, as a consequence, to the under-drying of individual layers of the seed mass.

To reduce the thermal injury and increase the germination of seeds it is necessary to limit the influence of temperature or even prevent it, and to ensure uniform drying of all seeds. These conditions are satisfied by the technology of drying seeds using vacuum. With this technology, the moisture from the seeds is removed by reducing the partial pressure of water vapor in the external environment. This technology has proven itself in the food industry [1-5].

Nevertheless, researches of the effect of vacuum drying of seed grains and industrial crops were insignificant, making it impossible to assess the availability of this technology. Therefore, these researches are relevant and necessary for improving the quality of post-harvest seed treatment.

2. Preconditions and means for resolving the problem

2.1. Analysis of recent research and publications

Researches [6-7] found that for drying without traumatizing seeds, the heating temperature of most seeds of crops should not exceed 40 ° C to 45 ° C. At the same time, the moisture content in one technological cycle of drying does not exceed 5%-6%, since with a larger value of the moisture content, cracks in the seeds are formed [8]. It was found that with the increase in the initial moisture content of the seeds, the maximum permissible temperature of seed heating falls [8]. To prevent injury to seeds when drying has been proposed to use mild temperature conditions convection drying [9]. However, when using soft temperature regimes of presence with undried seeds. At the same time, existing heat generators do not always provide the necessary stable temperature regime.

In [10] it was proposed to reduce the energy consumption for heating and drying the periodic evacuating rapeseed. To smooth the moisture content of the rape seed, a vibrating surface was applied, which adversely affects the quality of the seeds.

Researches [11] found that to increase the intensity of moisture separation, it is necessary, under vacuuming, to apply constant heating of seeds to a temperature close to the permissible temperature. Other researchers propose to use the vacuum-pulse method to increase drying speed [12-13]. However, using these methods could be a risk of thermal injury to the seed.

Existing researches have established critical temperatures for heating and the rate of moisture separation during drying of seeds, but it is not known what regimes of vacuum drying of seeds provide a rapid moisture release at parameters (heating temperature) that are less than permissible.

2.2. Purpose of the research

Search for options of intensive vacuum drying of seeds of grain crops with parameters, it is unacceptable to heat traumatize the seeds.

3. Results and discussion

To research the vacuum drying regimes, a simple experimental setup was developed and manufactured, which makes it possible to vacuum the seeds of grain crops to a pressure of 2 kPa (Fig. 1). The seeds were heated by an external heater to a temperature of 30 ° C. Electronic weights and a moisture meter were used to calculate the moisture release and the drying rate.

The drying speed v_W was calculated using the formula:

$$v_W = \frac{\Delta m}{m_0 \cdot \tau_E} \cdot 100\%, \quad (1)$$

where Δm is the change in weight of the sample during drying during exposure time τ_E , g;

m_0 — initial weight of the sample, g.

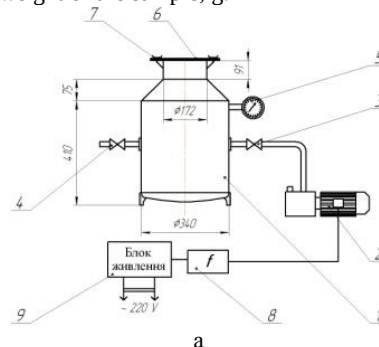


Fig. 1. – The construction scheme (a) and general view (b) of the experimental plant for the study of vacuum drying of seeds of crops: 1 – vacuum chamber; 2 – rotary vane vacuum pump subzero to the vacuum; 3 – faucet of a vacuum pump; 4 – faucet for throwing down of vacuum; 5 – vacuum-gauge; 6 – cover with gasket; 7 – locking mechanism; 8 – frequency converter; 12 – power supply.

The object of drying was a sample of maize seeds weighing 490 g and an initial humidity of 20%, which was placed in a vacuum chamber, in which various modes of changing the air pressure were used. The minimum air pressure in the vacuum chamber, which was used in the studies, was 4 kPa.

In direct evacuation weighed corn seeds at a temperature 19 °C were placed in a vacuum chamber at a pressure of 4 kPa for 75 minutes, after which the pressure was restored to atmospheric pressure (fig. 2).

As a result, the moisture separation and drying rate were calculated, which for corn were approximately equal to 2.6 g and 0.006% / min, respectively.

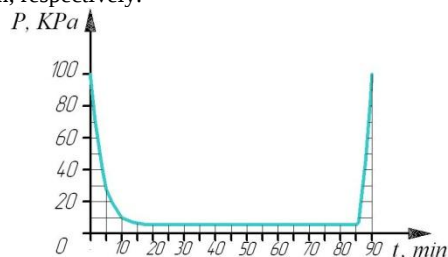


Fig. 2. – Change in pressure in the environment during the direct evacuation of grain seeds

The rate of vacuum drying was significantly lower than convective drying. Since the samples of the seeds were dried in containers, water vapor, which was released during drying, filled the intergranular space in the absence of air movement, which worsened the conditions of moisture separation. Therefore, the sample of corn seeds was filled up in a flat vessel in one layer, while its weight was 190 g. The rate of drying of maize seeds in one layer increased, respectively, to 0.025% / min. The moisture release was 4 g.

For the purpose of determining effect of air movement (blowing) on the drying speed, the regime was used with partial blowing of the seeds and increasing the pressure to 60 KPa — drying time 115 min (Fig. 3a) and a blowdown mode for 10 min at 40 KPa — drying time 70 min (Fig. 3b).

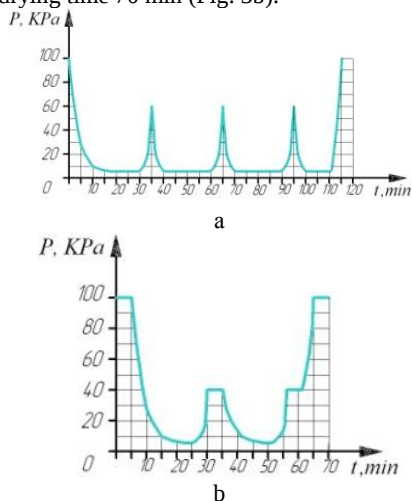


Fig. 3. — The pressure change in the surrounding space by direct evacuation seeds of crops: a — partial blowing mode of seeds and increased pressure to 60 kPa; b — blowdown mode for 10 min at 40 KPa

As in the previous regime, we used a sample of maize seeds weighing 490 g, an initial humidity of 20%, and a temperature of 19 °C.

The following results were obtained: for the regime (Fig.3a) — moisture extraction: 3 g, drying rate: 0.005% / min; for the mode (Fig.3b) — moisture extraction: 6 g, drying speed: 0.017%/min.

The drying speed of the regime (Fig.3a) turned out to be at the direct vacuum level, since during the blowing time, which is a pressure pulse with an amplitude of 60 kPa and a duration of 10 min, the decrease in the concentration of water vapor in the near-grain space is insufficient. When the blowing mode is at a fixed pressure (Fig.3b), the drying speed is half the amount of the direct

evacuation mode in one layer, in spite of blowing at constant pressure (40 kPa).

For all the above process modes seed drying in vacuo there was them cooled to 10 °C. The decrease in temperature during drying is due to the consumption of thermal energy of the seed for the evaporation of water. That is, drying in a vacuum is an adiabatic process. As the pressure in the near-grain space decreases, intense evaporation of moisture from the surface of the seed takes place, and as a consequence, its cooling, which leads to a slowing down of the diffusion of internal moisture outwards. This explains the low drying rate compared to convection. To increase drying speed, periodic heating of the seeds during the drying process is necessary.

To verify these claims, the regime was tested with periodic heating of the seeds to a temperature of 33 °C, evacuation to a pressure of 4 kPa and blowing for 5 minutes at atmospheric pressure. For this linkage used also maize seeds weighing 490 gr of preliminary heated to a temperature of 30 °C. The graphs of temperature and ambient pressure (Figure 4).

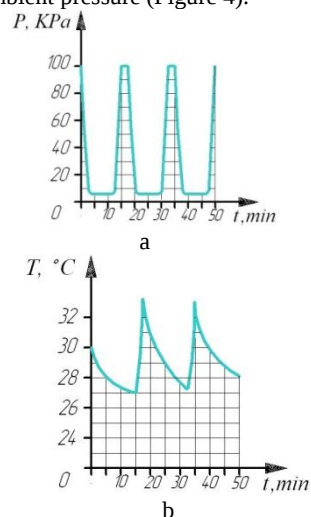


Fig. 4. – The change in the ambient pressure (a) and the heating temperature of the seeds (b) during the periodic heating and evacuation of seeds of cereals

In carrying out this mode, dehumidification drying maize seeds was observed in vacuo at 51 g and the drying speed was 0.259%/min, which is significantly higher than in previous modes. This confirmed the allegations put forward earlier. By varying the depth of the vacuum and the temperature of the seed heating, the rate of drying can be adjusted. Since the heating of seeds is carried out below the critical temperature, then the risk of thermal injury is minimal. Therefore, the application of this mode of drying grain seeds is promising. Further studies of this regimen with regard to the influence on the ability to germinate and the growth force of seeds of agricultural crops will improve the quality of seed material and reduce the thermal costs of drying.

4. Conclusions

As a result of experimental researches by vacuum drying of cereal seeds by the example of maize seeds it was found that the most intense dehumidification (drying rate 0.259% / min) without thermal injury observed during batch mode by heating, evacuation and purging at atmospheric pressure. Intensive dehumidification occurs due to compensation of heat losses in evaporation in a vacuum, which maintains a stable diffusion rate.

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METHOD FOR DIAGNOSING THE TIGHTNESS OF MILK PIPES MILKING INSTALLATIONS

СПОСОБ ДИАГНОСТИРОВАНИЯ ГЕРМЕТИЧНОСТИ МОЛОКОПРОВОДОВ ДОИЛЬНЫХ УСТАНОВОК

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Abstract: Checking the tightness of the milk lines of the milking plants with the help of indicators suggests comparing the results of the indicator readings when the milk line is switched off and turned on. The tightness of the milk lines can also be determined from the rate of incidence of the vacuum pressure from 50 to 36 kPa after the vacuum system of the milking unit is disconnected from the vacuum pumps. In order to eliminate the shortcomings of one and the other methods, it is suggested to check the tightness of the milk pipeline or its individual sections by placing an indicator piston in it. When the piston moves along the milk pipeline, the time of its movement is fixed, the vacuum pressure in the pre-piston and plug-in spaces of the milk pipeline. According to these indicators, by calculation, the tightness of the milk pipe is determined (air flow through leaks). The obtained result is compared with the permissible value of the tightness and make a conclusion about the technical condition of the milk line. At the same time, the costs of diagnostics are reduced, the accuracy of the diagnosis is increased, and the integrity of individual sections and loops of the milk pipeline is checked.

KEYWORDS: MILK LINE, VACUUM, PRESSURE, TIGHTNESS, MILKING UNIT, VACUUM PUMP, DIAGNOSIS, TECHNICAL CONDITION

Аннотация: Проверка герметичности молокопроводов доильных установок с помощью индикаторов предполагает сравнение результатов показаний индикатора при отключенном и включенном в работу молокопроводе. Герметичность молокопроводов можно также установить и по скорости падения вакуумметрического давления с 50 до 36 кПа после отключения вакуумной системы доильной установки от вакуумных насосов. С целью исключения недостатков одного и другого способов, предлагается проверять герметичность молокопровода или его отдельных участков, поместив в него поршень-индикатор. При движении поршня по молокопроводу фиксируется время его движения, вакуумметрическое давление в предпоршневом и запоршневом пространствах молокопровода. По этим показателям, расчетным путем, определяется герметичность молокопровода (расход воздуха через неплотности). Полученный результат сравнивают с допустимой величиной герметичности и делают вывод о техническом состоянии молокопровода. При этом снижаются затраты на диагностирование, повышается точность диагностирования, обеспечивается проверка герметичности отдельных участков и петель молокопровода.

КЛЮЧЕВЫЕ СЛОВА: МОЛОКОПРОВОД, ВАКУУМ, ДАВЛЕНИЕ, ГЕРМЕТИЧНОСТЬ, ДОИЛЬНАЯ УСТАНОВКА, ВАКУУМНЫЙ НАСОС, ДИАГНОСТИРОВАНИЕ, ТЕХНИЧЕСКОЕ СОСТОЯНИЕ

1. Introduction / Введение

Значительным источником повышения эффективности доения считается соблюдение стереотипа [1,2,3,4,5]. Его нарушение, по мнению И.А. Барышникова [6], приводит к торможению условного рефлекса молокоотдачи, в результате которого 97% разового удоя должно перейти из альвеолярного отдела в цистерну вымени. Вторично же его вызвать, в одну и ту же дойку, как считают ученые - физиологи, невозможно. Систематическое несоблюдение стереотипа доения приводит к ухудшению состояния здоровья животного, качества продукции [1,3,5,7,8].

С технической точки зрения стереотип определяется статической и динамической стабильностью рабочих режимов доильного аппарата. Основными из них, по мнению многих исследователей, являются величина разрежения под соском, частота пульсаций, соотношение тактов, характер воздействия сосковой резины на сосок вымени и другие [4,5,7,8,9,10,11]. Их стабильность, определяющая качество функционирования исполнительных механизмов (доильных аппаратов), главным образом, зависит от качества функционирования молоковакуумной системы [5,8,12,13].

При большой протяженности вакуумных линий в коровниках и доильных помещениях трудно своевременно реагировать на изменение вакуума в системе [4,5,7,8,9,10,12]. Недостатком системы контроля состоит в том, что показывающие приборы применяются для параметров, изменение величин которых во времени носит вероятностный характер [5,9,11]. Своевременное и точное диагностирование герметичности молокопроводов позволит обеспечить статическую и динамическую стабильность рабочих режимов доильных установок в процессе доения.

2. Preconditions and means for resolving the Problem / Предпосылки и средства для решения проблемы

При проверке герметичности молокопроводов доильных установок с помощью индикаторов КИ-4840 сравнивают результаты показаний индикатора при отключенном и включенном в работу молокопроводе. Разность показаний пропорциональна количеству воздуха, попадающего в молокопровод за счет неплотностей в нем [5,8,9]. Этот способ имеет некоторые недостатки: большие затраты времени на проверку герметичности, сложность конструкции и дороговизна индикатора, ограничение пропускной способности величиной 70 м³/ч, невозможность диагностирования отдельных участков и петель молокопровода.

Герметичность молокопроводов также устанавливают и по скорости падения вакуумметрического давления с 50 кПа до 36 кПа после отключения вакуумной системы доильной установки от вакуумных насосов [5,8,9,11,12]. В число недостатков этого способа можно отнести: невозможность фиксации вакуумметрического давления на отметке 50 кПа из-за отсутствия быстродействующих задвижек, зависимость точности измерения от вместимости вакуумной системы, невозможность диагностирования отдельных участков и петель молокопровода.

3. Solution of the examined Problem / Решение исследуемой проблемы

Проверить герметичность молокопровода или его отдельных участков можно поместив в него поршень-

индикатор[5,14]. При движении поршня по молокопроводу фиксируется время его движения, вакуумметрическое давление в предпоршневом и запоршневом пространствах молокопровода. По этим показателям, расчетным путем определяется герметичность молокопровода (расход воздуха через неплотности). Полученный результат сравнивают с допустимой величиной герметичности и делают вывод о техническом состоянии молокопровода. При этом снижаются затраты на диагностирование, повышается точность диагностирования, обеспечивается проверка герметичности отдельных участков и петель молокопровода.

Для вывода формулы расчета герметичности представим отверстия неплотностей в виде одной узкой щели, направленной вдоль образующей цилиндрической трубы 1. Ширина этой щели y в общем случае равна $y = f(x)$. Левый торец закрыт (рисунок 1).

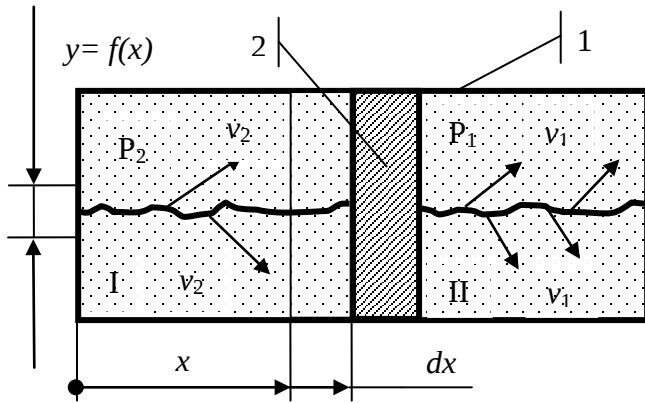


Рисунок 1 – Схема для расчёта герметичности молокопровода:

1 - молокопровод; 2 - поршень; v_1 - скорость атмосферного воздуха, натекающего в трубу через щель при давлении внутри трубы $P_1 < P_a$, м/с; v_2 - скорость атмосферного воздуха, натекающего в трубу через щель при давлении внутри трубы $P_2 < P_a$, м/с; I - предпоршневое пространство молокопровода; II - запоршневое пространство молокопровода; ширина щели $y = f(x)$ вдоль образующей цилиндрической трубы 1, м

В отсутствие поршня 2, при соответствующей подаче вакуумных насосов и настроен вакуумного регулятора, в трубе постоянно поддерживается рабочее давление P_1 , несмотря на наличие подсоса воздуха через щель. Если в этот момент в трубу поместить поршень 2, то в запоршневом пространстве II давление начнет повышаться до величины P_2 . При этом $P_a > P_2 > P_1$, где P_a - атмосферное давление. Поршень начинает двигаться по трубе в сторону меньшего давления P_1 . Движущей силой поршня будет сила F .

$$F = (P_1 - P_2)\pi R^2 > F_{тр}, \quad (1)$$

где $F_{тр}$ - сила трения, Н; R - внутренний радиус молокопровода, м.

В дальнейших рассуждениях приняты следующие допущения:

- сила трения по всей длине трубы постоянна;
- сила инерции $m \frac{d^2x}{dt^2}$ и сила трения воздуха, текущего по трубе вслед за поршнем, пренебрежительно малы по сравнению с силой трения $F_{тр}$, вследствие малости массы поршня и сравнительной малости скоростей движения поршня;
- прососы воздуха в зазоре между поршнем и стенкой трубы отсутствуют.

За время dt объем запоршневого пространства возрастает на величину $dV = \pi R^2 dx$ за счет того, что через щель длиной $(x + dx)$ и шириной $f(x)$ в него натекает атмосферный воздух. Объем этого воздуха будет равен

$$dV = [(x + dx) f(x)] v_2 dt, \quad (2)$$

где $(x + dx)f(x)$ - площадь щели, m^2 ; v_2 - скорость воздуха, натекающего через щель в запоршневое пространство, м/с.

Дифференциальное уравнение движение поршня-индикатора будет иметь вид

$$\pi R^2 dx = [(x + dx) f(x)] v_2 dt, \quad (3)$$

или

$$\pi R^2 dx = v_2 f(x)(x dt + dx dt). \quad (4)$$

Пренебрегая членом $dx dt$ как величиной второго порядка малости, получим зависимость

$$\pi R^2 dx = v_2 f(x) x dt. \quad (5)$$

Значение величины $y = f(x)$ в общем случае неопределимо. Заменим $f(x)$ приведенной шириной щели b , имеющей по всей длине трубы постоянное значение. Приведенной шириной щели b называется ширина такой щели, расход воздуха через которую равен действительному расходу Q_d через все реальные неплотности, т.е. $b l v_2 = Q_d$, откуда

$$b = \frac{Q_d}{l v_2}, \quad (6)$$

где l - длина трубы, м; b - приведенная ширина щели, м.

Преобразовав уравнение (5), получим соотношение

$$\frac{dx}{dt} = \frac{b v_2}{\pi R^2} x. \quad (7)$$

Ввиду постоянства величин v_2 , b , π , R , введём обозначение

$$\frac{b v_2}{\pi R^2} = B, \quad (8)$$

тогда

$$\frac{dx}{dt} = v_n = B x, \quad (9)$$

где v_n - скорость продвижения поршня-индикатора в молокопроводе, м/с.

Формула (9) показывает, что поршень-индикатор в трубе будет двигаться ускоренно со скоростью, линейно возрастающей с ростом x . Время T прохождения поршнем-индикатором расстояния от l_0 до l получим, проинтегрировав зависимость (7), предварительно разделив переменные [15].

$$\int_{l_0}^l \frac{dx}{x} = B \int_0^T dt. \quad (10)$$

После преобразования получим

$$BT = \ln \frac{l}{l_0}, \quad (11)$$

$$l = l_0 e^{BT}, \quad (12)$$

$$B = \frac{1}{T} \ln \frac{l}{l_0}. \quad (13)$$

В этих зависимостях l_0 всегда больше нуля. Если же $l_0 = 0$, то натекания атмосферного воздуха в запоршневое пространство не будет. Движения поршня происходить не будет.

Из формулы (11) следует, что если через молокопровод пропускать поршни с различной силой трения, но обязательно в пределах l_0, l , то $BT = \text{const}$.

Определим значение приведенной ширины щели b , используя формулы (8) и (13).

$$b = \pi R^2 \frac{1}{v_2} \cdot \frac{1}{T} \ln \frac{l}{l_0}. \quad (14)$$

Имея значение приведенной ширины щели и ее длину, можно определить расход воздуха через нее при любом давлении внутри трубы $P_1 < P_a$. Например, расход воздуха Q_1 через такую щель при рабочем давлении в молокопроводе P_1 будет равен

$$Q_1 = b l v_1, \quad (15)$$

где v_1 – скорость атмосферного воздуха, натекающего в трубу через щель при давлении внутри трубы $P_1 < P_a$ м/с; bl – площадь щели в молокопроводе, m^2 .

Подставив значение величины b из (14), получим,

$$Q_1 = \pi R^2 \frac{1}{T} \cdot \frac{v_1}{v_2} \ln \frac{l}{l_0}. \quad (16)$$

Выражение l/T в данной формуле представляет собой среднюю скорость поршня-индикатора $\bar{v}_n$ на участке l .

$$\bar{v}_n = \frac{1}{T}. \quad (17)$$

Ширина щели b в реальных условиях изменяется от нескольких единиц до сотен микрометров. Поэтому при определении скоростей v_1 и v_2 вполне правомерно использовать закономерности для истечения газов и жидкостей через малые отверстия [16,17,18].

На основании этих закономерностей

$$v_1 = \varphi \sqrt{2g\Delta P_1}, \quad (18)$$

$$v_2 = \varphi \sqrt{2g\Delta P_2}, \quad (19)$$

где $\Delta P_1 = (P_a - P_1)$; $\Delta P_2 = (P_a - P_2)$ – разность давлений в предпоршневом и запоршневом пространствах трубы и давлением воздуха на наружной поверхности трубы, Па; φ – коэффициент скорости.

Разность давлений ΔP_1 и ΔP_2 есть вакуумметрическое давление (вакуум), измеряемое вакуумметрами, установленными соответственно в предпоршневом и запоршневом пространствах молокопровода

$$\Delta P_1 = P_{b1}, \quad (20)$$

$$\Delta P_2 = P_{b2u}. \quad (21)$$

Подставив значение v_1 и v_2 из формул (18) и (19) и $\bar{v}_n$ из (16), после преобразований, получим

$$Q_1 = \pi R^2 \bar{v}_n \sqrt{\frac{P_{b1}}{P_{b2}}} \ln \frac{l}{l_0}, \quad (22)$$

либо

$$Q_1 = \pi R^2 \frac{1}{T} \sqrt{\frac{P_{b1}}{P_{b2}}} \ln \frac{l}{l_0}. \quad (23)$$

Для учета реальных условий движения поршня-индикатора (наличие перетечек воздуха через зазор между поршнем и трубой, непостоянство силы сопротивления по длине молокопровода [18,19]) результат, полученный по формуле (23), необходимо умножить на поправочный коэффициент « k », определяемый опытным путем. По данным авторов коэффициент $k=1,05...1,1$. В окончательном варианте зависимость (23) принимает вид

$$Q_1 = k \pi R^2 \frac{1}{T} \sqrt{\frac{P_{b1}}{P_{b2}}} \ln \frac{l}{l_0}. \quad (24)$$

4. Results and discussion, conclusion / Результаты и дискуссия, заключение

Таким образом, используя движение поршня-индикатора по молокопроводу с фиксацией времени его движения, вакуумметрического давления в предпоршневом и запоршневом пространствах молокопровода, расчетным путем определяется герметичность молокопровода (расход воздуха через неплотности). Полученный результат сравнивают с допустимой величиной герметичности и делают вывод о техническом состоянии молокопровода. При этом снижаются затраты на диагностирование, повышается точность диагностирования, обеспечивается проверка герметичности отдельных участков и петель молокопровода.

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ТЕХНОЛОГИЧЕСКИЕ И ТЕХНИЧЕСКИЕ ПРИНЦИПЫ ПОВЫШЕНИЯ КАЧЕСТВА РАБОТЫ КАРТОФЕЛЕУБОРОЧНЫХ МАШИН

TECHNOLOGICAL AND TECHNICAL PRINCIPLES FOR IMPROVING THE QUALITY OF POTATO HARVESTERS

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Abstract: *The main issues are considered in this topic: the mathematical model to determine the optimum parameter of potato harvester; analysis of the potato harvesters process as material balance equation; foundation of technological and technical principles for improving the quality of potato harvesters.*

KEYWORDS: POTATO, HARVESTING, POTATO HARVESTER, QUALITY, PARTS OF HARVESTER, PARAMETER

1. Вступление

Картофель является важной сельскохозяйственной культурой, урожай которой используется для питания людей, кормления животных и в качестве сырья в промышленности. При этом, картофелеводство является одной из наиболее трудо- и энергоемких отраслей сельского хозяйства. Существенные затраты энергии наблюдаются при выполнении всех технологических операций, особенно уборки. Поэтому, эффективность картофелеводства существенно зависит от использования техники в процессе выращивания и уборки урожая.

Заключительной операцией при выращивании картофеля, которая выполняется в полевых условиях, является уборка. Именно по качественным и количественным показателям уборки оценивают урожай и определяют его конкурентоспособность на рынке продукции.

Для уборки картофеля во всем мире используются навесные и полуприцепные картофелекопатели, а также прицепные и самоходные картофелеуборочные комбайны известных фирм-производителей GRIMME, AVR, LOCKWOOD и т.д., которые отличаются показателями технической характеристики и качеством выполнения процесса, хотя при этом сохраняется общий типаж рабочих органов, используемых в технологических схемах машин [1].

В процессе работы картофелекопателей последовательно выполняется выкапывание клубненосного шара почвы и просеивание мелкой почвы, а также укладывание других составляющих в валок на поверхность поля для последующего подбора (комбайнами или вручную). Но уровень механизации производственных процессов современного сельского хозяйства определяют целесообразность использования картофелеуборочных комбайнов, которые кроме названных операций картофелекопателей выполняют также отделение почвенных комков, растительных примесей и камней, с одновременной перегрузкой собранного урожая в транспортные средства или загрузкой в бункер комбайна [2].

2. Предпосылки и средства решения проблемы

Эффективность работы картофелеуборочных машин оценивают качественными (потери и повреждение клубней, а также содержание примесей) и эксплуатационными показателями (производительность, затраты топлива и другие).

Как показывает анализ исследований и испытаний картофелеуборочных машин, на показатели эффективности работы существенно влияют условия уборки и характеристики культуры [3], которые на одном и том же поля могут варьироваться в широких пределах. Но независимо от этого, при уборке картофеля уборочными машинами следует достичь

качественные показатели, которые соответствуют агротехническим требованиям.

Существенное влияние на качественные показатели урожая картофеля оказывают: технологии возделывания и уборки, выбор техники и правильная организация уборочного процесса в целом [4]. Во многих странах используется технология компании GRIMME, которая при правильной реализации технологии позволяет получать высокий и качественный урожай картофеля.

На основании анализа использования современной картофелеуборочной техники в реальных производственных условиях возникла необходимость в формировании технологических и технических принципов, которые могли бы обеспечить высокое качество выполнения механизированной уборки картофеля.

Вопросам исследования процесса уборки картофеля уборочными машинами и совершенствования картофелеуборочных машин посвящены работы многих ученых.

Сред основных направлений совершенствования картофелеуборочных машин можно выделить [5-15]:

- разработка и совершенствование картофелеуборочной техники для работы при разных почвенно-климатических условиях [2,5,6,7,10,15];

- совершенствование рабочих органов уборочных машин (прежде всего, сепарирующих) на основании системных решений. При совершенствовании сепарирующих рабочих органов особое внимание уделяется интенсификации просеивания почвы. Поскольку качество выполнения процесса рабочими органами зависит от условий работы, разрабатываются также рекомендации по применению сменных элементов конструкции уборочных машин [2,5,6,7,8,10,13,14,15];

- повышение производительности надежности уборочных машин с целью выполнения механизированной уборки картофеля в оптимальные сроки [8,10,12,15];

- снижение повреждения и потерь клубней при механизированной уборке на основании разработки схем уборочных машин с минимальными перепадами клубней картофеля при движении по поверхности рабочих органов [2,9,10,11,15] и другие.

В данной работе приводится анализ функционирования картофелеуборочных машин и поиск путей повышения качества их работы с учетом влияние эффективности отдельных ее рабочих органов на основании уравнения материального баланса [16].

3. Решение рассматриваемой проблемы

Процесс уборки картофеля картофелеуборочными машинами представим в виде системы, которая состоит из убираемой культуры и непосредственно уборочной машины. Факторами

системы являются характеристики культуры картофеля $\{R\}$, погоднo-климатические условия $\{P\}$, конструктивно-кинематические $\{S\}$ и эксплуатационные параметры уборочной машины $\{K\}$.

При этом могут быть выделены как наиболее значимые факторы: урожайность картофеля U , влажность почвы w , глубина подкапывания h и рабочая скорость движения уборочной машины V . В зависимости от указанных факторов определяют подачу клубненосной подкопанной массы почвы на рабочие органы q и производительность машины W .

Основными качественными показателями работы картофелеуборочных машин является величина общих потерь Y_1 и повреждений клубней Y_2 , а также содержание примесей Y_3 (в относительных массовых единицах), которые контролируются в процессе работы.

Для упрощения задачи оптимизации параметров может быть использован обобщенный мультипликативный критерий оптимизации вида [17]

$$(1) Y = Y_1 \cdot Y_2 \cdot Y_3,$$

или с учетом степени значимости отдельных критериев

$$(2) Y = Y_1^a \cdot Y_2^b \cdot Y_3^c,$$

где a, b, c – показатели степени, определяющие значимость соответствующих критериев. На основании экспертной оценки установлено: $a = 0,54, b = 0,31, c = 0,15$.

Качественная работа картофелеуборочной машины достигается решением задачи параметрической оптимизации в виде системы

$$(3) \begin{cases} (u_0 - u) \rightarrow \min, \\ (g_0 - \Delta g) \rightarrow \min, \\ (p_0 - \Delta p) \rightarrow \min, \\ (r_0 - \Delta r) \rightarrow \min, \end{cases}$$

где u_0, g_0, p_0, r_0 – подача на сепарирующие рабочие органы уборочной машины в картофельном ворохе клубней, мелкой почвы, почвенных комков и растительных примесей соответственно; $\Delta g, \Delta p, \Delta r$ – интенсивность просеивания мелкой почвы, отделения почвенных комков и растительных примесей; u – интенсивность выгрузки картофеля из машины.

Согласно уравнения материального баланса [16]

$$(4) q_0 = u_0 + g_0 + p_0 + r_0,$$

где q_0 – подача картофельного вороха на рабочие органы уборочной машины, которая зависит от скорости движения и ширины захвата уборочной машины, глубины подкапывания и т.д.

Для определения подачи воспользуемся корреляционной зависимостью, полученной в результате обработки графических зависимостей [2]

$$(5) q_0 = (73,4175 + 41,7879V^2 - 113,293V - 5,8381h + 0,4708h^2 + 11,1924Vh) \cdot i,$$

где i – количество одновременно убираемых машиной рядков картофеля.

Подачу клубней определим согласно выражения

$$(6) u_0 = 0,01 \cdot U \cdot B \cdot V,$$

где B – ширина захвата уборочной машины.

Существенной составляющей картофельного вороха (до 90...95%) являются почвенные примеси (как мелкой, так и крупной фракций) [2]. Именно от работы подкапывающих и сепарирующих рабочих органов картофелеуборочных машин, а также качества выполнения предварительной обработки почвы, зависит содержание почвенных примесей в собранном урожае клубней. Почвенные комки (в том же числе, и непроходной фракции) формируются в нижних плотных слоях почвы и их разрушение достигается вследствие продолжительного действия циклической нагрузки или под действием комбинации подкапывания пласта, его разрушения и сепарации почвы с минимальным повреждением клубней.

Одним из важных условий эффективной работы картофелеуборочной машины является обеспечения оптимальной подачи картофельного вороха q_{opt} , при котором достигается высокое качество урожая

$$(7) q_0 \rightarrow q_{opt}.$$

Значение оптимальной подачи q_{opt} зависит как от свойств картофельного вороха, так и от технологической схемы машины, параметров и режимов работы рабочих органов.

В оптимизационную модель вводятся также ограничения по глубине подкапывания h , рабочей скорости движения уборочной машины V , влажности почвы w и других факторах

$$(8) \begin{cases} h \geq h_{max}, \\ V_{max} \geq V \geq V_{min}, \\ w_{max} \geq w \geq w_{min}, \\ \dots \end{cases}$$

Поскольку потери клубней, просеивания мелкой почвы, разрушение и отделение почвенных комков, а также отделение растительных примесей возможно на соответствующих рабочих органах картофелеуборочных машин, составляющие системы (3) запишем в виде

$$(9) \begin{cases} (u_0 - u) = \Delta u_I + \Delta u_{II}, \\ \Delta g = \Delta g_I + \Delta g_{II} + \Delta p_I, \\ \Delta p = \Delta p_I + \Delta p_{II}, \\ \Delta r = \Delta r_{II}. \end{cases}$$

Компьютерное моделирование на основании описанных оптимизационных моделей с помощью ПП MathCAD и сравнение результатов с экспериментальными данными доказывают целесообразность их практического применения.

4. Результаты и дискуссии

На основании проведенного системного анализа системы (3) с учетом выражений (4-9) предложены основные технологические и технические принципы повышения качества функционирования картофелеуборочных машин, среди которых можно выделить следующие.

Существенное (на 45...63%) повышение сепарирующей способности и производительности (на 27...50%) уборочных машин достигается путем использования в технологических схемах картофелеуборочных машин рабочих органов с комбинациями: подкапывание + разрушение почвенного пласта (активные, пассивные или комбинированные); подкапывание + разрушение почвенного пласта + просеивания почвы (активные, пассивные или комбинированные).

Особое внимание следует обращать на соответствие условий уборки установленным диапазонам условий работы уборочных машин, поскольку современная уборочная техника ведущих фирм-производителей обеспечивает высокое качество выполнения процесса только при благоприятных условиях работы. В случае повышения влажности или твердости почвы будет наблюдаться снижение сепарирующей способности просеивающих сепараторов и повышение уровня повреждений клубней.

Для адаптации режимов работы рабочих органов уборочных машин к условиям работы целесообразно использовать систему контроля и оперативного регулирования параметров и режимов работы машин, оборудованную бортовым компьютером и навигационным оборудованием. На основании существующих методов возможным является построение прогностической карты урожайности, которая загружается в бортовой компьютер и является основанием для формирования управляющего сигнала для исполнительных механизмов. С учетом времени запаздывания срабатывают исполнительные механизмы и обеспечивают необходимое регулирование параметров и режимов работы уборочной машины. Для снижения влияния ошибки прогнозирования урожайности производится постоянное ее измерение и правка прогностической карты.

Среди фундаментальных основ повышения эффективности использования картофелеуборочных машин можно выделить разработку научных основ проектирования рабочих органов и адаптации их параметров и режимов работы к условиям уборки на основании математического моделирования и системного анализа его результатов.

5. Заключение

Процесс уборки картофеля картофелеуборочными машинами является сложной динамической системой, которая характеризуется изменением подачи клубненосной массы и содержания в ней составляющих: клубней и примесей. Интенсивность изменения содержания примесей в массе является стохастической величиной и зависит от характеристик условий уборки, параметров и режимов работы уборочной машины и других факторов.

Повышение качества работы картофелеуборочных машин можно достичь путем решения многокритериальной задачи оптимизации, основными критериями которой являются потери и повреждения клубней, содержание в ворохе мелкой почвы, почвенных комков и растительных примесей с учетом ограничений по условиям работы уборочных машин.

На основании проведенного системного анализа полученной оптимизационной модели обоснованы основные технологические и технические принципы повышения качества функционирования картофелеуборочных машин: использование в схеме уборочных машин комбинированных рабочих органов, которые позволяют повысить сепарирующую способность рабочих органов и содержание примесей в ворохе; соответствие технологий возделывания и уборки почвенно-климатическим условиям, а также выбор техники и организация уборочного процесса согласно этих технологий; использование уборочной техники только при рекомендованных фирмами-производителями условиях; возможность адаптации уборочной машины с использованием прогностических карт урожайности и с учетом параметров биосреды.

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МАТЕМАТИЧЕСКОЕ МОДЕЛИРОВАНИЕ ПРОСТРАНСТВЕННОГО ДВИЖЕНИЯ МОТОАГРЕГАТА С УЧАСТИЕМ ОПЕРАТОРА

MATHEMATICAL MODELING OF SPATIAL MOVEMENT OF WALKING TRACTOR WITH THE PARTICIPATION OF THE OPERATOR

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Резюме: Разработана динамическая модель пространственного движения мотоагрегата с учетом динамических процессов, возникающих при движении. Представленная модель позволяет определить основные параметры движения и влияние внешних возмущений на оператора, а также дает возможность оценить динамические параметры агрегата в процессе конструирования и автоматизации процессов управления движением.

КЛЮЧЕВЫЕ СЛОВА: МОТОАГРЕГАТ, ДИНАМИЧЕСКАЯ СИСТЕМА, МАТЕМАТИЧЕСКОЕ МОДЕЛИРОВАНИЕ, УРАВНЕНИЕ ЛАГРАНЖА, ПРОСТРАНСТВЕННОЕ ДВИЖЕНИЕ.

Abstract: A dynamic model of the spatial motion of the motor unit with dynamic processes occurring during the motion is developed. The presented model allows to determine the main parameters of motion and the influence of external disturbances on the operator, and also makes it possible to evaluate the dynamic parameters of the unit in the process of designing and automating motion control processes.

KEY WORDS: WALKING TRACTOR AGGREGATE, DYNAMIC SYSTEM MODELING PROCESSES, LAGRANGE EQUATION, SPATIAL MOTION.

1. Введение

Мотоорудия и мотоагрегаты (МА) находят все большее применение в аграрном секторе (личные подсобные хозяйства, приусадебные и дачные участки, фермерские участки и т.п.), коммунальном и лесном хозяйствах, строительстве, там, где использование традиционных агрегатов затруднено или ограничено. Работа МА возможна только при участии оператора. Причем оператор не только управляет движением, но и участвует в тяговой динамике агрегата, что приводит к его быстрой утомляемости и снижению производительности агрегата в целом [1]. Установление причин, побуждающих оператора принимать физическое участие в тяговой динамике агрегата, оценка интенсивности и величины работы целесообразно провести с помощью математической модели движения МА.

2. Предпосылки и средства решения проблемы

Вопросам математического моделирования движения машинно-тракторных агрегатов посвящено достаточно много работ известных ученых. Однако в этих работах рассматривается движение автономных агрегатов, где оператор выполняет только управляющие функции через органы управления. Поэтому, предложенные математические модели не приемлемы для описания функционирования МА.

Проведен анализ большого числа работ [2, 3, 4, 5, 6, 7, 8], посвященных взаимодействию движителей с почвой, на основе чего определены наиболее приемлемые зависимости.

Достаточно большое количество работ посвящены оценке пространственного перемещения машин и агрегатов [9], однако большинство из них громоздки и требуют наличия сложных бортовых вычислительных систем. Необходимо отметить, что ни одна из теорий не рассматривает непосредственное взаимодействие оператора в динамике движения агрегата.

В работе [1] отмечается, что функционирование МА существенно отличается от условий работы традиционных машинно-тракторных агрегатов. Поэтому, функционирование МА необходимо рассматривать как систему оператор-мотоагрегат-среда, что обуславливает определенные взаимосвязи между подсистемами, таких как оператор-мотоагрегат, оператор-среда, мотоагрегат-среда и т.п.

Для описания взаимодействия почвообрабатывающих орудий использовались уточненные зависимости, предложен-

ные В.П. Горячкиным [3], а применительно к МА - описанная в работе [8]. Рассмотрение процессов взаимодействия движителей с опорным основанием использовались модели, предложенные В.В. Гуськовым и Хейдекем [2, 3], а также оригинальная методика, предложенная Горячкиным В.П. и развитая в дальнейшем Лопаревым [7] и Овсянниковым С.И. [8]. Однако данная методика требует дальнейшего экспериментального подтверждения.

Взаимодействие оператора с МА рассматривали Келлер Н.Д. [10], Горин Г.С., Овсянников С.И. [1] В этих работах отмечается, что оператор МА прикладывает значительные физические усилия, значение которых в несколько раз превышают допустимые. Следует отметить, что в этих работах рассматривается взаимодействие как результат воздействия машины на человека и не рассматриваются причины их возникновения.

Таким образом, процессы взаимодействия оператора и МА являются основополагающими в повышении производительности МА и требуют дальнейшего изучения.

Поэтому, целью данной работы является разработка обобщенной пространственной математической модели движения МА учитывающей взаимодействие с оператором при различных режимах и условиях эксплуатации. Применение математической модели позволит оценить физическое взаимодействие оператора и МА с целью поиска более эффективных средств снижения физического участия оператора при работе с МА.

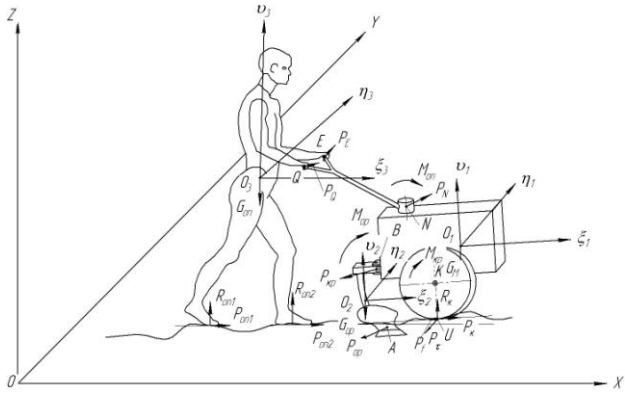
3. Результаты и дискуссия

Общая пространственная схема движения МА, управляемая оператором, представлена на фиг. 1. Для удобства описания и выполнения расчетов рассмотрим частные случаи динамических процессов в горизонтальной (фиг. 2), продольно-вертикальной (фиг. 3) и поперечно-вертикальной (фиг. 4) плоскостях.

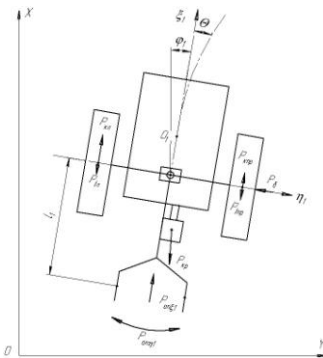
В математической модели приняты следующие допущения. Пространственное движение орудия заменено соответствующими силами, передаваемыми от орудия на остов мотоблока. Пространственное движение оператора заменено силами и моментами, передаваемыми от оператора на мотоблок через штанги управления.

Для преобразования перемещений подвижной системы координат $o_1\xi_1\eta_1\nu_1$ относительно неподвижной $oxyz$ использо-

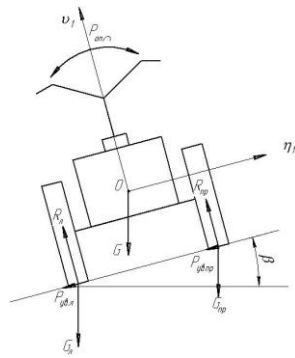
вался метод управляющих косинусов. Динамика движения описывалась общим уравнением Лагранжа.



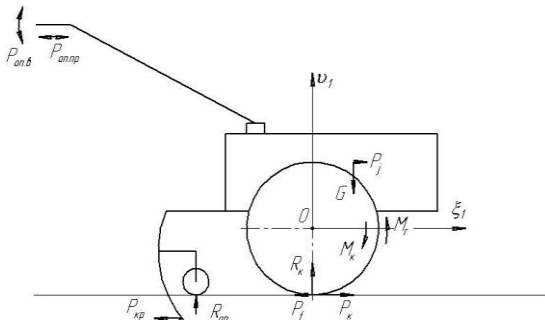
Фиг. 1. Пространственная схема к динамической модели движения МА.



Фиг. 2. Силы, действующие в горизонтальной плоскости.



Фиг. 3. Силы, действующие в вертикально-поперечной плоскости.



Фиг. 4. Силы, действующие в вертикально-продольной плоскости.

3.1 Силы, действующие на МА.

В горизонтальной плоскости действуют силы: касательные силы тяги, развиваемые движителями P_{K1}, P_{K2} ; силы сопротивления движению P_{f1}, P_{f2} ; силы бокового увода $P_{б}$; силы на крюке $P_{кр}$; усилия, прикладываемые оператором в продольном $P_{он\xi_1}$ и поперечном $P_{он\eta_1}$ направлениях.

Для определения касательных сил тяги и силы сопротивления качению грунта, воспользуемся моделью Гуськова В.В. [2]. Нормальная нагрузка G_i на i -ое колесо определялась методом расчета перераспределения веса мотоблока по бортам под действием гравитационных и инерционных сил. Силы сопротивления качению пневматической шины определим по модели, предложенной Хейдекем [3]. Тяговое сопротивление орудия $P_{ор}$ и нагрузка на крюке $P_{кр}$ определялась в зависимости от параметров технологического оборудования и физико-механических свойств почвы по уточненной формуле В.П. Горячкина [4]. В продольном направлении оператор прилагает

усилия только в случаях нарушения тягового баланса агрегата, выраженного в недостаточности сцепных свойств и повышенном буксовании. В поперечном направлении оператор прикладывает усилия для изменения направления курсового движения. Величина их зависит от наличия (отсутствие) дифференциала трансмиссии, сцепных свойств движителей, длины штанг управления и др. факторов. Момент управления определяется для каждого типа МА отдельно.

В вертикально-поперечной плоскости действуют силы веса МА G_a и орудия $G_{ор}$, реакции опорного основания на движители R_{np} и R_l и орудие $R_{ор}$, силы бокового увода $P_{бnp}$ и $P_{бл}$ под действием поперечного крена, усилий оператора на штанги управления $P_{он\eta}$ от поперечного крена на опрокидывание. Силы и реакции определяются и рассчитываются в зависимости от угла крена и действий моментов инерции [8].

В вертикально-продольной плоскости дополнительно к перечисленным действуют реактивный момент M_r численно равный крутящему моменту $M_{кр}$ на оси колес и противоположно направленный, усилия оператора в вертикальной плоскости $P_{овв}$, направленные на изменение вертикального положения рабочих органов орудия.

3.2 Кинетическая энергия

Кинетическая энергия системы определялась как сумма кинематической энергии поступательного T_n и вращательного T_e движений (1). Принята гипотеза об отсутствии диссипации энергии внутри системы $\frac{\partial T}{\partial q_i} = 0$, тогда уравнение Лагранжа

примет вид (2), а первый член запишем с учетом действующих сил и перемещений на основные оси системы (3):

$$T = T_n + T_e = \frac{m}{2} (\dot{x}_1^2 + \dot{y}_1^2) + \frac{J_1}{2} \dot{\varphi}_1^2 \quad (1)$$

$$\frac{d}{dt} \left(\frac{\partial T}{\partial \dot{q}_i} \right) + \frac{\partial \tilde{I}}{\partial q_i} = Q_i \quad (2)$$

$$\frac{d}{dt} \left(\frac{\partial T}{\partial \dot{\xi}_1} \right) = m \ddot{\xi}_1 + m \dot{\varphi}_1^2 l_1 \cos \varphi_1 + m \ddot{\varphi}_1 l_1 \sin \varphi_1$$

$$\frac{d}{dt} \left(\frac{\partial T}{\partial \dot{\eta}_1} \right) = m \ddot{\eta}_1 - m \dot{\varphi}_1^2 l_1 \sin \varphi_1 + m \ddot{\varphi}_1 l_1 \cos \varphi_1$$

$$\begin{aligned} \frac{d}{dt} \left(\frac{\partial T}{\partial \dot{\varphi}_1} \right) &= m \ddot{\xi}_1 \sin \varphi_1 l_1 + m \ddot{\eta}_1 \cos \varphi_1 l_1 + m \dot{\varphi}_1^2 l_1^2 - \dot{\varphi}_1 \dot{\eta}_1 m l_1 \sin \varphi_1 + \\ &+ \dot{\varphi}_1 \dot{\xi}_1 m l_1 \cos \varphi_1 + J \ddot{\varphi}_1 = m \ddot{\xi}_1 \sin \varphi_1 l_1 + m \ddot{\eta}_1 \cos \varphi_1 l_1 + \ddot{\varphi}_1 (m l_1^2 + J_1) + \\ &+ (\dot{\xi}_1 m l_1 \cos \varphi_1 - \dot{\eta}_1 m l_1 \sin \varphi_1) \dot{\varphi}_1 \end{aligned} \quad (3)$$

3.3 Математическая модель

Перемещение МА происходит под действием неконсервативных сил. Для определения обобщенных сил Q_i составим выражение элементарных работ внешних сил на возможных перемещениях. При этом коэффициенты при вариациях обобщенных координат являются обобщенными силами:

$$dA = dA_{\xi_1} + dA_{\eta_1} + dA_{\varphi_1}, \quad (4)$$

где

$$dA_{\xi_1} = \left(P_{ке} \cdot \cos \varphi_1 + P_{кв} \cdot \cos \varphi_1 - P_{fe} \cdot \cos \varphi_1 - P_{fv} \cdot \cos \varphi_1 - \right. \\ \left. - P_A \cdot \sin \varphi_1 - D_{ев} \cdot \cos \varphi_1 - D_{ев} \cdot \sin \varphi_1 \right) d\xi_1; \quad (5)$$

$$dA_{\eta_1} = \left(P_{ке} \sin \varphi_1 + P_{кв} \sin \varphi_1 - P_{fe} \sin \varphi_1 - P_{fv} \sin \varphi_1 + P_A \cos \varphi_1 - \right. \\ \left. - D_{ев} \sin \varphi_1 + D_{ев} \cos \varphi_1 \right) d\eta_1 \quad (6)$$

$$dA_{\varphi_1} = \left(P_{ке} \cdot \frac{B}{2} - P_{fv} \cdot \frac{B}{2} - P_{ке} \cdot \frac{B}{2} + P_{fv} \cdot \frac{B}{2} - D_{ев} \cdot l_k \right) d\varphi_1 \quad (7)$$

Если учесть, что $Q_i = \frac{\partial A_i}{\partial q_i}$, то получим

$$\begin{aligned} Q_{E1} &= (P_{k1} + P_{k2} - P_{f1} - P_{f2} - R_x) \cos \varphi_1 - (P_\delta + R_y) \sin \varphi_1 \\ Q_{\eta 1} &= (P_{k1} + P_{k2} - P_{f1} - P_{f2} - R_x) \sin \varphi_1 + (P_\delta + R_y) \cos \varphi_1 \\ Q_{\varphi 1} &= (P_{k2} - P_{k1} + P_{f1} - P_{f2}) \cdot \frac{B}{2} - R_y l_k \end{aligned} \quad (8)$$

С учетом выражений (3) и (8), а также, что в горизонтальной плоскости $\frac{\partial \Pi}{\partial q_i} = 0$, получим систему уравнений, описывающую движение мотоблока в горизонтальной плоскости:

$$\begin{cases} m \ddot{E}_1 + m \dot{\varphi}_1^2 l_1 \cos \varphi_1 + m \ddot{\varphi}_1 l_1 \sin \varphi_1 = \\ = (P_{k1} + P_{k2} - P_{f1} - P_{f2} - R_x) \cos \varphi_1 - (P_\delta + R_y) \sin \varphi_1 \\ m \ddot{\eta}_1 - m \dot{\varphi}_1^2 l_1 \sin \varphi_1 + m \ddot{\varphi}_1 l_1 \cos \varphi_1 = \\ = (P_{k1} + P_{k2} - P_{f1} - P_{f2} - R_x) \sin \varphi_1 + (P_\delta + R_y) \cos \varphi_1 \\ m \ddot{E}_1 \sin \varphi_1 l_1 + m \ddot{\eta}_1 \cos \varphi_1 l_1 + m \dot{\varphi}_1^2 l_1^2 - \dot{\varphi}_1 \dot{\eta}_1 m l_1 \sin \varphi_1 + \\ + \dot{\varphi}_1 \dot{E}_1 m l_1 \cos \varphi_1 + J_1 \ddot{\varphi}_1 = (P_{k2} - P_{k1} + P_{f1} - P_{f2}) \frac{B}{2} - R_y l_k \end{cases} \quad (9)$$

Обозначим:

$$\begin{aligned} C_1 &= m \dot{\varphi}_1^2 l_1 \cos \varphi_1; \\ D_1 &= (P_{k1} + P_{k2} - P_{f1} - P_{f2} - R_x) \cos \varphi_1 - (P_\delta + R_y) \sin \varphi_1; \\ B_2 &= m l_1 \cos \varphi_1; \\ C_2 &= -m \dot{\varphi}_1^2 l_1 \sin \varphi_1; \\ D_2 &= (P_{k1} + P_{k2} - P_{f1} - P_{f2} - R_x) \sin \varphi_1 + (P_\delta + R_y) \cos \varphi_1; \\ A_3 &= m l_1 \sin \varphi_1 + m l_1 \cos \varphi_1; \\ B_3 &= m l_1^2 + J_1; \\ C_3 &= (\dot{E}_1 m l_1 \cos \varphi_1 - \dot{\eta}_1 m l_1 \sin \varphi_1) \dot{\varphi}_1; \\ D_3 &= (P_{k2} - P_{k1} + P_{f1} - P_{f2}) \frac{B}{2} - R_y l_k; \end{aligned} \quad (10)$$

После выполнения преобразований система дифференциальных уравнений (9) примет вид:

$$\begin{cases} m \ddot{E}_1 + B_1 \ddot{\varphi}_1 + C_1 = D_1 \\ m \ddot{\eta}_1 + B_2 \ddot{\varphi}_1 + C_2 = D_2 \\ A_3 (\ddot{E}_1 + \ddot{\eta}_1) + B_3 \ddot{\varphi}_1 + C_3 = D_3 \end{cases} \quad (11)$$

Решим данную систему аналитически относительно $\ddot{E}_1, \ddot{\eta}_1, \ddot{\varphi}_1$. Тогда по методу Крамера:

$$\ddot{E} = \frac{\Delta}{\Delta_{\ddot{E}}}, \quad \ddot{\eta} = \frac{\Delta}{\Delta_{\ddot{\eta}}}, \quad \ddot{\varphi}_1 = \frac{\Delta}{\Delta_{\ddot{\varphi}_1}} \quad (12)$$

Система линейных дифференциальных уравнений (11) решается методом Рунге-Кутты 4-го порядка с предварительным ее приведением к виду Коши [9].

Аналогичным образом определялись системы линейных дифференциальных уравнений для плоскостей вертикально-поперечной (13) и вертикально-продольной (14).

$$\begin{cases} m \ddot{\eta}_2 + B_1 \ddot{\varphi}_2 + C_1 = D_1 \\ m \ddot{v}_2 + B_2 \ddot{\varphi}_2 + C_2 = D_2 \\ A_3 (\ddot{\eta}_2 + \ddot{v}_2) + B_3 \ddot{\varphi}_2 + C_3 = D_3 \end{cases} \quad (13)$$

$$\begin{cases} m \ddot{\xi}_3 + B_1 \ddot{\varphi}_3 + C_1 = D_1 \\ m \ddot{v}_3 + B_2 \ddot{\varphi}_3 + C_2 = D_2 \\ A_3 (\ddot{\xi}_3 + \ddot{v}_3) + B_3 \ddot{\varphi}_3 + C_3 = D_3 \end{cases} \quad (14)$$

Силловые уравнивающие действия оператора в общем случае можно разложить на три составляющие по осям глобальной системы координат $Ronx, Rony, Ronz$. Данные величины являются компонентами главного вектора Ron , который вместе с главным моментом Mon определяет силовое участие оператора в работе МА. Главные момент Mon также можно разложить на три составляющие. Момент $Monx$ – определяет противодействие оператора боковому опрокидыванию МА. Момент $Mony$ определяет противодействие оператора повороту МА в продольной плоскости под действием реактивного момента на оси колеса, а также при заглоблении (выглоблении) рабочих органов орудия. Момент $Monz$ – управляющий момент, характеризующий курсовое движение МА.

Для определения главного вектора Ron и главного момента Mon составим дифференциальное уравнение движения в трех взаимно ортогональных плоскостях.

В плоскости XY (координата x):

$$\begin{aligned} m \cdot \ddot{x} &= \sum R_x; m \ddot{x} = P_{k1} + P_{k2} - P_{f1} - P_{f2} - R_x \pm R_{\ddot{v}0} \\ J_x \ddot{\varphi} &= \sum M_x; J_x \ddot{\varphi}_1 = P_d \cdot r_k - R_{\ddot{v}y} \cdot h_o + M_{\ddot{v}0} \end{aligned} \quad (13)$$

В плоскости YZ (координата y):

$$\begin{aligned} m \ddot{y} &= \sum R_y; m \ddot{y} = -P_d \pm R_{\ddot{v}y} \\ J_y \ddot{\varphi}_2 &= \sum M_y; J_y \ddot{\varphi}_2 = +P_{k1} \cdot r_k + P_{k2} \cdot r_k - P_{f1} \cdot r_k - P_{f2} \cdot r_k - \\ &- R_{e\delta} \cdot b_{e\delta} + R_{\ddot{v}0} - R_{\ddot{v}z} \cdot b_o + M_{\ddot{v}y} \end{aligned} \quad (14)$$

В плоскости XZ (координата z):

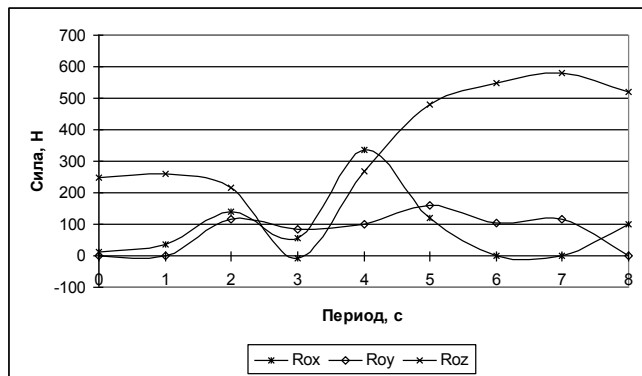
$$\begin{aligned} m \ddot{z} &= \sum R_z; m \ddot{z} = -mg + R_{\ddot{v}z} \\ J_z \cdot \ddot{\varphi}_3 : J_z \cdot \ddot{\varphi}_3 &= P_{k1} \frac{a}{2} - P_{f2} \frac{a}{2} - P_{k2} \frac{a}{2} + P_{f1} \frac{a}{2} - \\ &- R_y \cdot b_{e\delta} - R_{\ddot{v}y} \cdot b_o + M_{\ddot{v}z} \end{aligned} \quad (15)$$

Решая данное уравнение относительно неизвестных, получим:

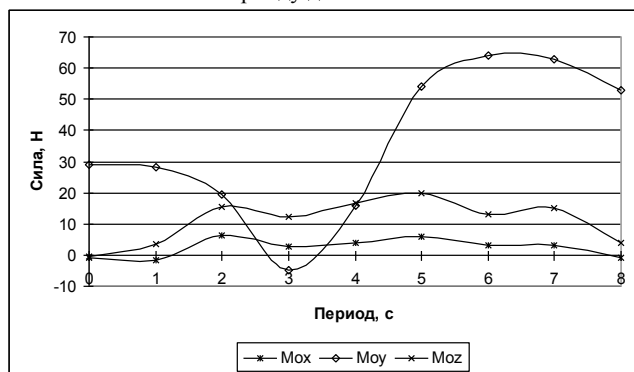
$$\begin{cases} R_{\ddot{v}0} = m \cdot \ddot{x} - P_{k1} - P_{k2} + P_{f1} + P_{f2} + R_{e\delta} \\ R_{\ddot{v}0} = m \cdot \ddot{y} + P_d \\ R_{\ddot{v}z} = m \cdot \ddot{z} + m \cdot g \\ M_{\ddot{v}0} = J_x \cdot \ddot{\varphi}_1 - P_d \cdot r_k + R_{\ddot{v}y} \cdot h_o \\ M_{\ddot{v}y} = J_y \cdot \ddot{\varphi}_2 - P_{k1} \cdot r_k - P_{k2} \cdot r_k + P_{f1} \cdot r_k + P_{f2} \cdot r_k + \\ + R_x \cdot b_{e\delta} - R_{\ddot{v}0} \cdot h_o + R_{\ddot{v}z} \cdot b_o = \\ = J_y \cdot \ddot{\varphi}_2 - P_d \cdot r_k + P_f \cdot r_k - R_x \cdot b_{e\delta} - R_{\ddot{v}0} \cdot h_o + R_{\ddot{v}z} \cdot b_o \\ M_{\ddot{v}z} = J_z \cdot \ddot{\varphi}_3 - (P_{k2} - P_{f2} - P_{k1} + P_{f1}) \cdot \frac{a}{2} + R_y \cdot b_{e\delta} + R_{\ddot{v}0} \cdot b_o \end{cases} \quad (16)$$

4 Результаты и дискуссия

Адекватность модели проверялась на моделировании динамических процессов при движении МА на базе мотоблока «Нева» по полю после уборки картофеля. Полученные спектральные характеристики сил и моментов представлены на фиг. 5 и 6.



Фиг. 5. Спектральная плотность распределения составляющих реакции силы на штангах управления МА по величине и периоду действия.



Фиг. 4. Спектральная плотность распределения составляющих главного момента на штангах управления МА по величине и периоду действия.

5 Заключение

Из полученных данных можно сделать вывод, что максимальное усилие оператор прикладывает в вертикальной плоскости, магнитуда которой достигает 600 Н с частотой воздействия до 7 сек. Модуль силы имеет положительное значение, что соответствует удержанию остова МА от опрокидывания под действием реактивного момента [12]. В продольном направлении оператор прикладывать толкающее усилие магнитудой до 320 Н с периодом 4 сек. В поперечном направлении оператор прикладывает усилия при выводе агрегата на заданную траекторию при уводе. Среди реактивных моментов наибольшее значение имеет момент относительно поперечной оси M_{oy} , параллельной оси колес направленный на удержание МА от бокового крена. Магнитуда момента достигает 65 Нм, а период изменения 6 - 7 сек.

В целом можно сделать вывод, что разработанная математическая модель пространственного движения мотоагрегата позволяет определить силовое взаимодействие оператора с МА в процессе движения.

Подтверждение

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CONTEMPORARY TRENDS IN THE IMPROVEMENT OF BEET TOP REMOVING MECHANISMS AND MACHINES

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Abstract: *There are considered contemporary trends in the improvement of the technological process for the preparation of sugar beet roots for digging out and directions of the development of the designs of the working tools for the removal of the tops. Analysis of contemporary machines has been carried out according to the indicators of their quality, energy consumption and efficiency of the technological process. Tasks have been determined for creating new working tools. It is noted that at the current stage of development of the beet top removing machines the efficiency and quality problems of the top separation process are solved by increasing the number of operations, which is an energy-consuming, material-intensive and costly way to improve the technological process and the working tools. And, as a consequence, it is necessary to look for technical solutions, technologically allowing operations required only for the preparation of the root crops to be harvested. Therefore, further research should be conducted in the direction of creating a working tool that combines good tracing of the heads by cleaners with low-energy intensity of the cutting tools.*

KEYWORDS: ROOT CROP, HEAD OF THE ROOT CROP, ROOT, SEPARATION OF ROOTS, WORKING TOOL.

1. Introduction

Increased pollution of the heap of root crops with the green mass by 1% reduces the yield of sucrose by 0.1%, and, when storing the root crops in open heaps (в караракс) with the top content of about 4%, the daily sugar losses are, on average, 0.012% [1]. For mechanised harvesting of sugar beet roots crops, there is a wide range of sugar beet harvesters and beet harvesting complexes that have both advantages and disadvantages. Especially these shortcomings are present in the preparation process of the root crops for digging, which includes the operations of separating the tops from root crops and removing it from the area of action of the digging tools or attaching to the haulm properties which do not affect the technological process of digging and separating the heap of root crops from impurities. The working tools for separation of the tops perform satisfactorily the technological process at relatively lower speeds in comparison with the digging tools.

Analysis of a number of designs of the working tools for the removal of the tops has been made in considerable detail in the works by L.V.Pogorelov, V.M.Bulgakov, M.V.Tatyanko, R.B.Gevko, V.Ya.Martinenko, N.M.Helemendik, etc., yet creation of efficient working tools that would be characterised by small energy intensity, high efficiency and simplicity of design remains an actual production and scientific task.

The most complete overview of the designs, technological schemes and working tools for the removal of the tops is given in work [1]. There is the history of creation and development of designs considered, the quality indicators of the technological process performed by the main working tools are analyzed, but there are no indicators of energy intensity.

In [2], a classification of the working tools for the separation of the tops is presented. In [3, 4, 5, 6] there is analysis of various technological schemes for the separation of the tops, the results of comparative tests of machines from the West European firms Holmer, Ropa, Kleine, Matrot, Grimme, TIM-Thyregood.

It should be noted that the assessment of the quality indicators of the top separation process according to the European standard is a visual assessment and therefore it is impossible to estimate the mass content of the tops in the root crop heap left behind different machines. The drive power is given only for individual machines.

The purpose of the work is to analyse contemporary technologies and technical means for removing sugar beet tops and to identify the most promising designs and directions for their improvement.

2. Results and discussion

Let us trace the main trends in the development of the working tools for the top removal using as an example modern sugar beet harvesters and top-separating modules of the leading domestic and foreign manufacturers. As a result of analysis, it has been established that the following operations can be carried out at the present stage when the top is removed [1,3,4,5,6]:

- cutting of the main mass of the tops on the root crop which is growing;
- removal of the basic mass of the tops by shock interactions (defoliation);
- disintegration of the tops and their distribution in rows or inter-rows;
- transportation of the tops to the vehicle or to the harvested field;
- partial disintegration of the plant residues in the inter-row spaces;
- cutting the root crop heads with the residues of leaves which remained after cutting the bulk of the tops;
- cleaning of the root crop heads from the residues of the foliage.

In order to analyse various technological schemes of contemporary machines and to determine the current trends in the development of the designs, we classify the technological process of preparing root crops for digging according to the above-mentioned operations (Fig. 1).

Cutting of the basic mass of the tops can be carried out with tracing of the root crop heads or without their tracing at a fixed height relative to the soil surface. In this case the following mechanisms are predominately used: rotors with a horizontal, vertical or inclined at an angle to the horizon axes of rotation.

Removal of the tops by shock interactions takes place with the help of rotors having flexible elements - defoliators.

The distribution of the tops in the inter-row spacing occurs simultaneously with the cutting and disintegration processes, by applying guiding surfaces in the housing of the cutting rotor. There is also known a practice of pushing the plant residues into the inter-row soil.

For transportation of the tops, augers, rod conveyors or a set of vertical augers are predominantly used.

Removal of the plant residues from the inter-row zone is carried out using the fan effect at rotary drums with a horizontal axis of rotation.

Cleansing of the root crop heads from the top residues is effected by rotors with flexible cleaning tools. The most common of these are blade cleaners.

Table 1. Analysis of technological schemes and working tools of the machines made by the world's leading companies for the top removal

Removal		The top removal operations / a design used for the execution of this technological operation							
N o.	Stages of top removal	BM-6B [1]	Kleine SF10, Moreau, Matrot, Holmer, Gilles [1,6,12,10,14]	Ropa Euro-Tiger V8-3 [6]	Grimme Maxtron 620 [13]	Ropa, Holmer, Gilles [5,10,13]	WIC, Grimme BM 330, Aloway Industries [7,3,8]	Garford Victor, MGSh6 (TKZ) [1]	WIC (Amity Technology) [9]
1	Removal of the basic mass of the tops	Cutting with tracing	Cutting without tracing						Top removal by shock interactions
		Active knife	Horizontal cutting rotor					Active knife with vertical shaft	Rotor with flexible blades
2	Elimination of the influence of the basic mass of the tops	Transportation of the tops into the vehicle or the harvested field	Disintegration and transportation to the harvested field	Disintegration and distribution in inter-rows	Disintegration and distribution in inter-rows with compaction	Disintegration and distribution in inter-rows and rows		Transportation to the harvested field	Disintegration and distribution in inter-rows and rows
		Rod elevators	Horizontal auger	Horizontal cutting rotor and guiding surfaces	Horizontal cutting rotor and rollers for pressing the tops into the soil	Horizontal cutting rotor		Vertical auger working tools	Horizontal rotor defoliator
3	Removal of the top residues	Primary cleaning of the root crop heads	Primary cutting of the heads with tracing					Primary cleaning	Primary cleaning with primary cutting
		Blade cleaner of the root crop heads	Passive knife			Passive knife	Passive or active knife	Primary cleaner with a hard surface	Two horizontal rotors-defoliators. Active knife or passive knife.
4	Elimination of the influence of the top residues	Transportation of residues to the harvested field	—	—	—	—	Disintegration and distribution in inter-rows	Disintegration and distribution in rows and inter-rows	Disintegration and distribution in inter-rows
		Blade cleaner of the root crop heads					Rotor from working tools sectionally, arranged above the rows	Primary cleaner with a hard surface	Rotor from working tools sectionally, arranged above the rows
5	Separation of the tops on the digging-separating working tools	Separation of individual particles of the bound and loose tops from the heap	Partial removal of the bound and detachment of the free disintegrated tops						
Required power, kW		44 - full power	—	—	—	—	75...90	24.7 (drive power)	75...90
Working speed, m·s ⁻¹		up to 1.5	up to 1.8					—	up to 1.8

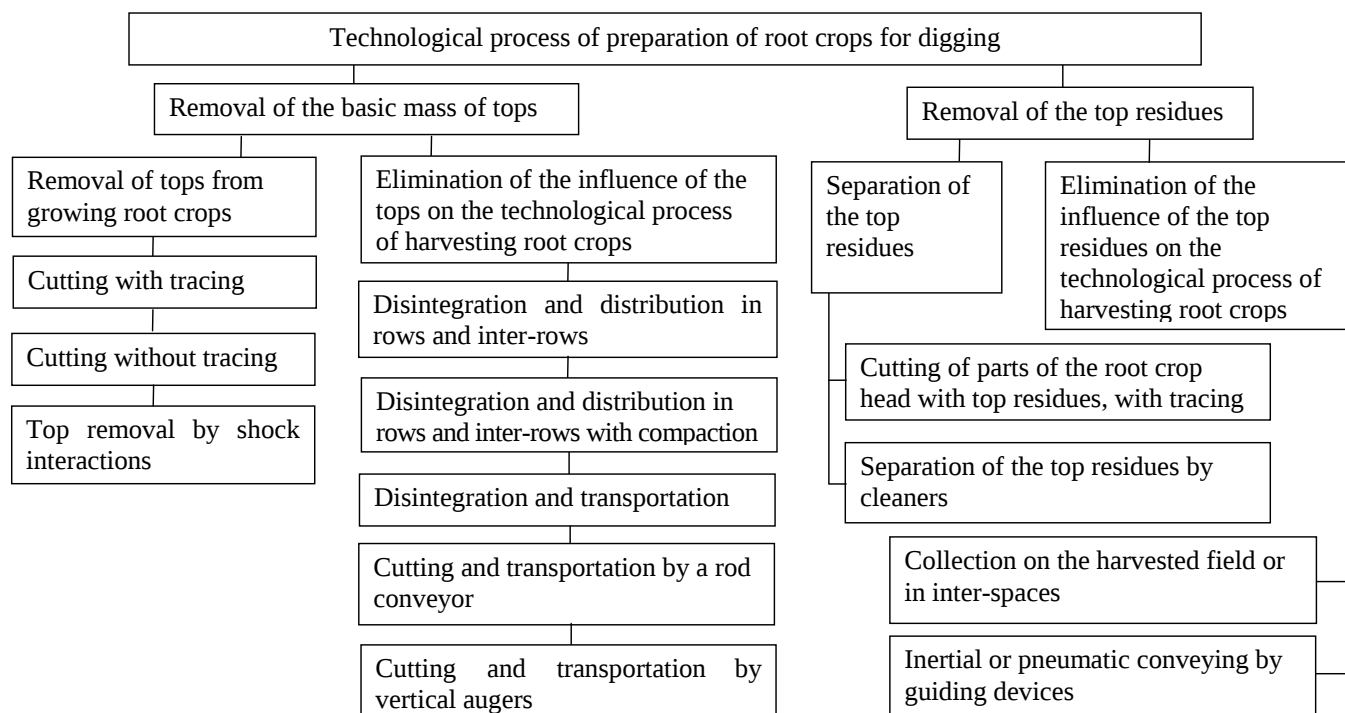


Fig.1. – Classification of the technological process of the root crop preparation for digging

On the basis of this classification let us analyse application of the aforementioned operations of the top removal in technological schemes of machines of the world's leading firms (Table 1). Also, we will display with the help of which constructions of working bodies these operations are implemented. We will also show by means of what designs of the working tools these operations are implemented.

Table 1 clearly manifests the tendency of the leading world's firms Holmer, Ropa, Kleine, Moreau et al. (Table 1) to apply cutting of the tops without using a tracer by means of a rotor with a horizontal axis of rotation, and using a tracer when cutting the root crop heads with passive cutters (Fig. 2). In individual combine harvesters a slightly modified technological scheme for the top removal is used. Instead of arcuate knives on the rotary top cutter drum, knives of various shapes are used, which allow the tops to be disintegrated and left in rows and inter-rows. In addition, the top is disintegrated to such a degree that it does not affect the work of the digging and separating tools of the root crop harvesting machines.

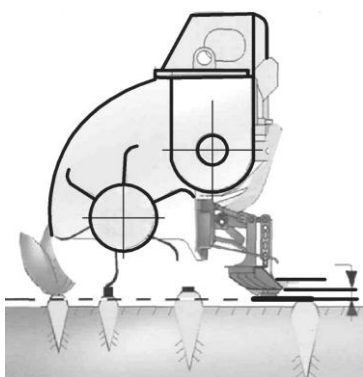


Fig. 2. – A top removing module without tracing and with passive tracing for primary cutting of the root crop heads

An original technical solution for the top separation was used by the firm Grimme in its Maxtron 620 combine harvester [3]. In it, the tops are cut only above the rows of sectionally arranged arc-shaped knives (Fig. 3a). On the shaft of the rotary top cutter, in the inter-row region, there are placed polyurethane beaters which are by 30 mm longer than the knives. Due to the suction effect, created by the long beaters, the plant residues are selected from the row region and fed into the region of the knives for disintegration. By means of the guiding flaps, located on the housing, the shredded tops are distributed in the inter-rows and pressed into the soil by steel rollers (Fig. 3b).

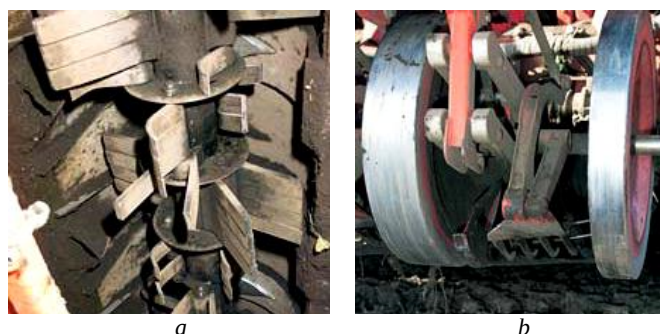


Fig. 3. – Peculiarities of the Maxtron 620 combine harvester: *a* – the working tools for the top separation; *b* – rollers for pressing the disintegrated tops into the ground

The next operation is primary cutting of the root crop heads with tracing, using passive primary cutters. Remote regulation of the cutting height is carried out by hydraulic cylinders, the top cutter having no supporting wheels. In order to control and automatically maintain the given cut-off height of the rotary top cutter, sensor signals are used from the primary cutter of the root crop heads.

Using the Harford Victor combine harvester (Fig. 4) as an example, it is possible to consider implementation of this

technological scheme with the help of fundamentally different working tools. Cutting of the tops also occurs without tracing by means of active knives with vertical-auger transporting tools, but cleaning the root crop heads from the top residues takes place by active-type cleaners with a rigid parabolic surface.

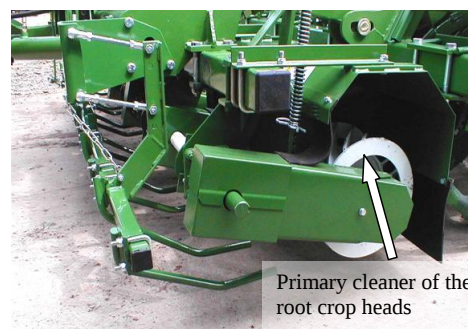


Fig. 4. – Cleaning device of the combine harvester Garford Victor

Grimme also puts out Rootster 604 trailed root crop harvesters. They work in a complex with the BM 330/300 top-harvesting machine. In this machine the technological process of separating the tops takes place in three stages (Fig. 5).

At first, the tops are cut off by a rotary top cutter with metal arc-shaped knives. The knives are arranged on the rotor shaft along a helical line (Fig. 6a). The next operation is cleaning of the root crop heads from the residues of the uncut tops by two shafts that rotate in opposite directions.

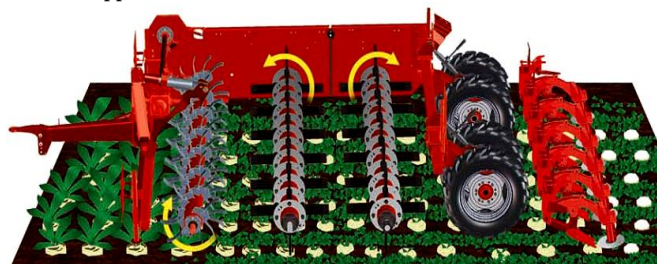


Fig. 5. – A flow chart of the Grimme BM 330 top-harvesting machine

On the shafts there are pivotally fixed polyurethane beaters (Fig. 6b). After cleaning the root crops are additionally cut. The cutting-off apparatus, which is used here for primary cutting of the root crop heads (Figure 6c) consists of a comb-shaped tracer and an active disk knife.

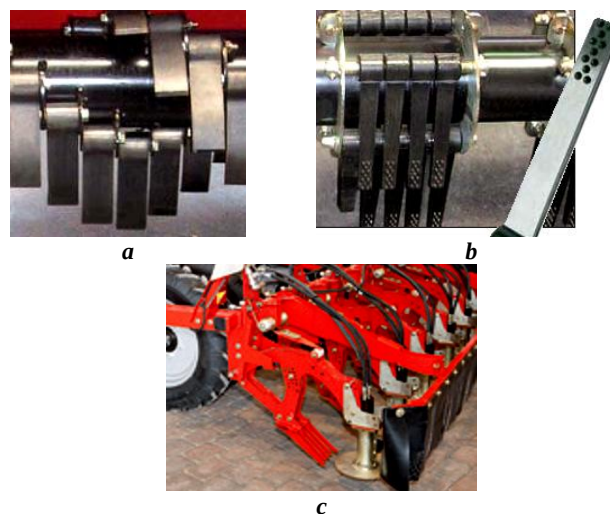


Fig. 6. – The working tools of the top harvesting machine BM 330: *a* – a rotary top cutter; *b* – counter-rotating polyurethane beaters on the shafts; *c* – a primary cutter of the active type with tracing

The WIT top harvesting machine from the Amity Technology (Fig. 7) [7] has an original design of the working parts. Instead of the traditional rotary top cutter a three-shaft beater-disintegrator (defoliation) is used in it. The working tools (beaters) are made from rubber of high mechanical strength. The working part of the beaters contains metal inserts, which increases the impact force of the beaters when they come into contact with the tops. Due to the high intensity of the action of the beaters the tops are crushed into small-size particles and sprayed into the inter-rows without adversely affecting the operation of the root crop harvesting machine.

The final completion of the top cutting process of the root crops is carried out by passive primary cutting knives moving above each of the rows and having an individual system for tracing the height of the root crop heads.

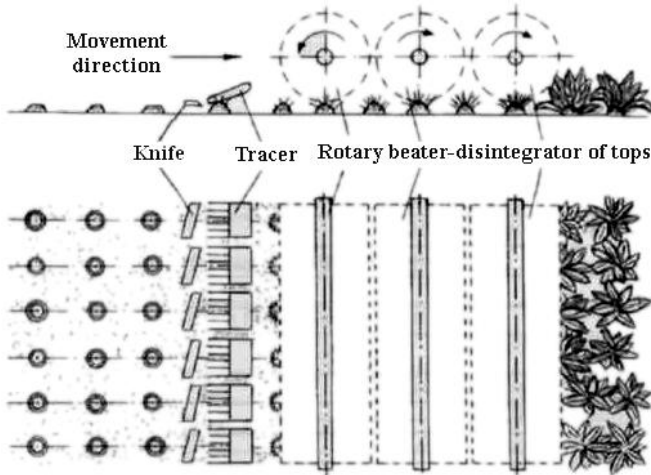


Fig. 7. – A flow chart of the WIT top harvesting machine from the Amity Technology

Analysing Table 1, we can draw a conclusion that the basic mass of the tops are removed:

- by cutting the tops without tracing, with primary cutting of the root crop heads with tracing (Fig. 8);
- by cutting the tops with tracing and cleaning the root crop heads from the top residues (Fig. 9);
- by shock interactions (Fig.10).

The technological scheme (Fig.8) is characterised by the complexity of the structures that are used, high metal intensity, and high energy costs due to the use of rotors with a horizontal and vertical arrangement of the rotation axis.

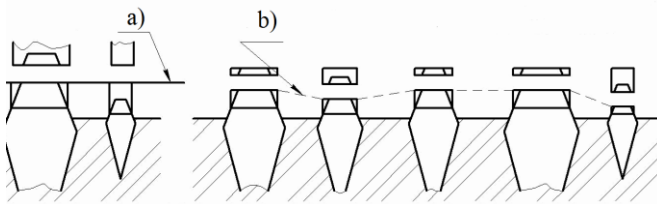


Fig. 8. A technological scheme of separation of the tops by cutting without a tracer (a); by primary cutting of the top residues from the heads with tracing (b)

Due to the application of cutting without tracing, the quality of the top removal in this technological scheme is insufficient, and it is possible that the sugar-bearing mass in the removed tops exceeds the agrotechnical requirements. The forward speed in this scheme is not greater than $1.8 \text{ m} \cdot \text{s}^{-1}$ since the working tools for the passive primary cutting of the root crop heads knock them out and form bevelled surfaces when the speed increases. It should also be noted that irregularities of the soil cause oscillations of the machine. In combination with varying heights of the root crop heads relative to the soil, this significantly affects the location of the cutting plane relative to each head of the root crop, which will lead to different thickness of the layer of the tops of the tops on the root crop heads (Fig. 8), worsening the tracing conditions, and, accordingly,

lowering the cutting quality. Due to the increased thickness of the head, the quantity of the top residues on the root crops will diminish, yet the loss of the sacchariferous mass will increase.

According to the results of independent tests [6], the number of root crops with a height of the tops higher than 2 cm was on average 2% for the machines of this technological scheme, there were 26.6% of the high-cut root crops, 7.5% of the low-cut root crops, but satisfactorily cut root crops - 60.1% %. Such statistics indicate the losses of the sacchariferous mass and a significant amount of tops left on the root crops. These results are given for an average working speed of the machines $6 \text{ km} / \text{h}$. This is a small speed of movement, and its choice is determined by a desire to achieve a quality of the passive primary cutters as satisfactory as possible. The digging tools can work at a much higher speed. Therefore the use of the passive primary cutters in this flow chart allows reduction of the residues of tops on the root crops after cutting without tracing by a rotary top cutting apparatus, yet the efficiency of the harvesting process of the root crops remains at a low level.

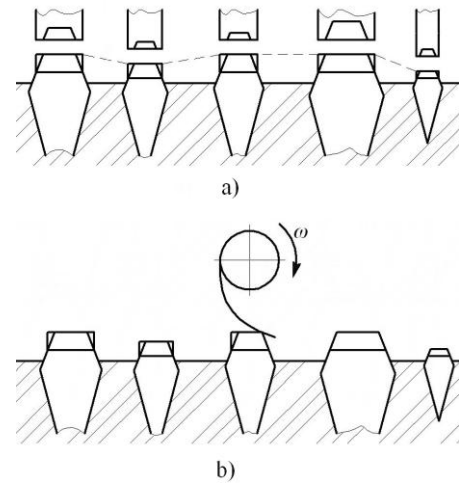


Fig. 9. – A technological scheme for the separation of the tops by cutting with tracing (a) and primary cleaning of the root crop heads by cleaners (b)

Thanks to cutting without tracing by means of a rotor with a vertical axis of rotation combined with an active-type cleaner (Garford Victor Harvester), the requirements for the separation quality of the tops can be more satisfied; however there may be root crops, damaged and kicked out of the soil. Considering that the required drive power of such an MGS-6 machine is 24.7 kW, a considerable amount of energy is wasted for cutting and transportation of the tops using a set of vertical augers.

A combination of cutting without tracing with a primary cutting of the root crop heads by means of active-type cutting apparatus of a small mass (Grimme BM 330 machine) provides a possibility for higher quality separation of the tops at working speeds up to $1.8 \text{ m} \cdot \text{s}^{-1}$, but the required power is 75 ... 90 kW. A characteristic feature of this design is an individual hydraulic drive for each knife, which made it possible to reduce the weight of the top cutter apparatus. But such a drive is rather expensive, which will definitely be reflected on the total cost of the machine.

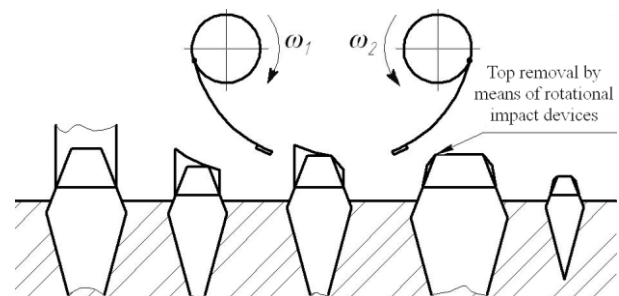


Fig. 10. – A technological scheme of the top separation of the tops by shock interactions

The technological scheme of cutting with tracing and primary cleaning of the root crop heads by a cleaner (BM-6B) is also characterised by the complexity of the design and the metal intensity through the need to use the tracing device in combination with the cutting tool. The quality of the top removal in such a technological scheme is much better; however, this time the forward speed of the machine is not high - up to $1.5 \text{ m} \cdot \text{s}^{-1}$. This is due to the fact that at higher speeds in the links of the tracing mechanism and in the cutting tool there arise inertial forces, and tracing of the heads deteriorates. The total power required to ensure the technological process is 44 kW.

Removal of the tops by shock interactions (separate models of the WIC machines) is characterised by significant energy losses due to additional disintegration of the tops and by the need to use three shafts with different directions of rotation. In this case the tops are disintegrated and distributed on the uncollected field. When the working speed is increased above $1.8 \text{ m} \cdot \text{s}^{-1}$, the quality of the top removal remains low. In order to carry out such a process, the required power is 75 ... 90 kW.

3. Conclusions

1. At the present stage of development of the top harvesting machinery the problems of efficiency and quality of the top separation process are solved by increasing the number of operations, which is an energy-consuming, material-intensive and costly way to improve the technological process and the working tools.

2. It is advised to carry out further research in the direction of creating a working tool that combines good tracing of the heads by cleaners with low-energy intensity of the cutting tools.

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EVALUATION OF THE HORIZONTAL AND VERTICAL FORCE OF REVERSIBLE MOLDBOARD PLOW IN SILTY LOAM SOIL.

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Abstract: The aim of the study was to evaluate the horizontal and vertical force of reversible moldboard plow at various operation depth and speed. A complete randomized block design was used with three replications, the first treatment was tillage depth with three levels 15, 20 25 cm and the second treatment was the forward speed with two levels 3.9, 6.3 km·h⁻¹. The result showed that both forces were increased as increased of tillage depth and speed, however, the operation depth was more effective than speed, on the other hand, the horizontal force was more affected than vertical force by increasing both of depth and speed.

KEY WORDS: VERTICAL, HORIZONTAL FORCE, MOLDBOARD PLOW, TILLAGE

1. Introduction

Tillage is a process that is created suitable soil conditions for seed germination and crop growth (Al-Suhaibani and Ghaly, 2010). Primary tillage in many cases is important for successful crop establishment, pest and weed control (Nielsen., et al 2017). It is the base operation in the agriculture and its energy represents a considerable part of the energy utilized in crop production (Askari& Khalifahamzehghasem, 2013). The most common implement in the world for plowing is a moldboard plow, ranging from the simplest type for animal traction, up to the highly advanced, large plows for tractors (Celik et al., 2007). It is widely used as a primary tillage tool (Mari et al., 2014). It was stated that draft force of moldboard plow was about 2.14 and 1.8 times as much as the chisel plow and disk harrow respectively (Askari &Khalifahamzehghasem, 2013). The amount of force required to pull or push the implement through the soil is called horizontal or draft force (Godwin, 2007).while the Vertical force of implement is assisting or preventing the penetration into the soil (Manuwa et al., 2012). Forward speed and tillage depth have considerable effect on the horizontal and vertical force. The vertical force increased with the increase in the tillage depth, this effect occurred due to the fact that at higher depths a greater soil volume is considered (Ibrahmi et al., 2014; Manuwa, 2009). It is clearly that tillage forces varied according to the shape of the tools, forward speed and depth of plowing (Godwin & Dogherty, 2007). The relationships between draft and speed showed that with increasing forward speed, draft increased (Safari & Gazor, 2014; Rosa & Wulfsohn, 2008). For the moldboard plow, when forward speed increased from 1.5 to 3 km·h⁻¹, the draft requirement increased from 9 to 10 kN and was not doubled. This increment occurred for all plows. In the sense of doubling the forward speed, the draft requirement increased but was not doubled (ASABE Standards, 2009). Also the ploughing depth has a greater effect on the horizontal force than on the vertical force (Al-Suhaibani & Ghaly, 2013).The objective of this study was to estimate the horizontal and vertical force of reversible moldboard plough in silty loam at different tillage forward speed and depth using new three hitch dynamometer (frame).

2. Materials and method

2.1 Treatments and Experimental Design

The trial was conducted at Czeslawice field in Lublin province (Poland) which belong to the farm of the University of Life Science in Lublin at a latitude of N51° 18' 23'', N51° 13' 27', the soil texture was classified as a silt loam.

A complete randomized block design was used with three replications, the first treatment was tillage depth with three levels 15, 20, and 25 cm, the second treatment was the forward speed with two levels 3.9, 6.3 km·h⁻¹

Statistical software (Statistica ver. 13.1) was used for the statistical analysis of the results obtained. A two-way analysis of variance (ANOVA) with interaction was performed assuming 0.05 as the level of significance

2.2. Forces measurements

The three-point hitch dynamometer (Figure 5) was built in the form of an inverted U-shape and calibrated at the same company (Industrial Institute of Agricultural Engineering, Poznan, Poland). It was designed and fabricated to be used for measuring the horizontal and vertical force of the soil engaged tools. It was placed between the tractor and the implement. The force sensing elements comprise five extended transducers. Moreover, the data acquisition system consists of a program data logger and serial cable. A notebook computer was also used for the monitoring and processing of data in the system.



Fig.1. Three hitch dynamometer (frame)

3. Result and discussion

3.1. Effect of tillage depth on the horizontal and vertical force

The results (Figure 2) indicated that change of tillage depth of plow was effective on the both horizontal and vertical force ($p < 0.05$).

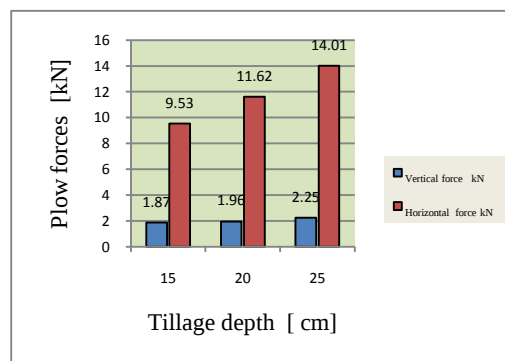


Fig. 2. Effect of tillage depth on the horizontal and vertical force

The maximum horizontal and vertical force (14.01 and 2.25 kN) occurred at tillage depth of 25 cm, while the minimum values (9.53 and 1.87) occurred at tillage depth of 15 cm. The forces increased with the operational depth of the plow was probably due to the greater depth of plow penetration, therefore, the larger surface area and greater soil resistance which requires more force to cut the soil and turn it over during plowing operation, this agreed with the findings of Deshpande et al. (2015)

3.2. Effect of forward speed on the horizontal and vertical force

The results (Figure 2) indicated that forward speed of plow was effective on the both horizontal and vertical force. The maximum horizontal and vertical force were occurred at forward speed 6.3 km·h⁻¹, and minimum values occurred at forward speed of 3.9 km·h⁻¹. these were agreed with finding of Manuwa (2009) who confirmed that horizontal force increased as forward speed increased, this is mainly due to the acceleration of the soil, which in turn, increased the reaction at the interface of tools, therefore, the higher sliding resistance

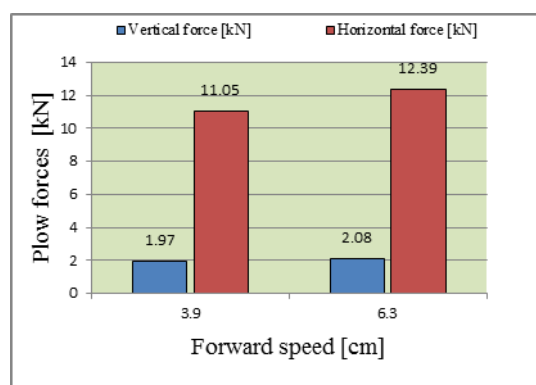


Fig 3. Effect of forward speed on the horizontal and vertical force

3.3. Effect of interaction of tillage depth forward speed on the horizontal and vertical force

The results (Figure 4) indicated that the interaction of change tillage depth and forward speed of moldboard plow was effective on the horizontal and vertical force the effect was significant at ($p < 0.05$). The highest horizontal and vertical force were obtained by forward speed of 6.3 km·h⁻¹ and tillage depth of 25 cm, while the minimum values were obtained by forward speed of 3.9 km·h⁻¹ and tillage depth of 15 cm.

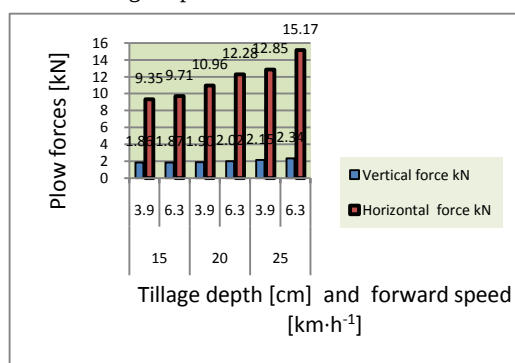


Fig 4. The interaction effect of tillage depth forward speed on the horizontal and vertical force

4. Conclusions

The horizontal and vertical forces of reversible moldboard plow are increased as forward speed and working depth increased. The effect of plowing depth was higher on the forces than effect of plowing speed. Moreover, the increasing in the horizontal force was higher than the increasing in vertical force.

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DETERMINATION OF THE OPTIMUM PARAMETERS OF THE ROLLERS OF THE ROLLER SURFACE FOR SORTING POTATOES.

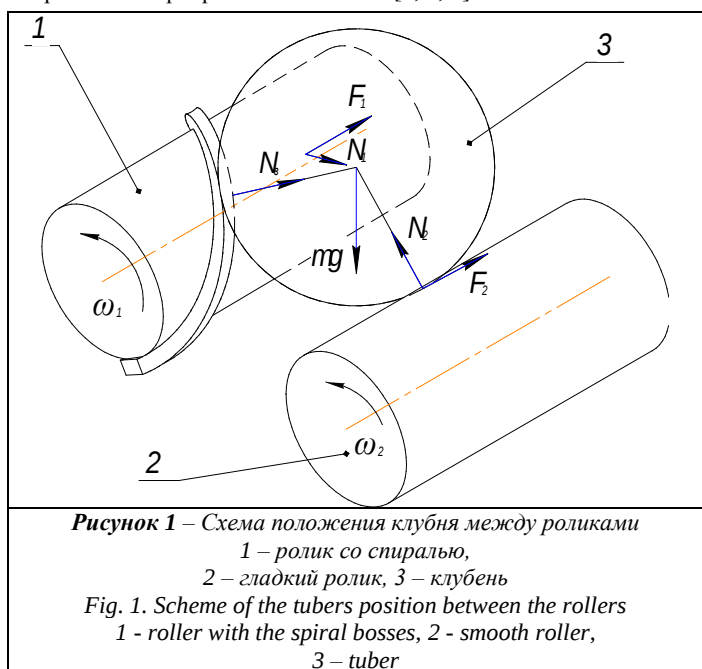
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Summary: The publication presents an analysis of operational and technical parameters affecting the performance of the roller surface in potato-calibrated machines.

KEYWORDS: SORTING OF POTATO, ROLLER SURFACE, WORKING AND TECHNICAL PARAMETERS.

Введение. Основным показателем работоспособности машины для калибровки клубней картофеля на размерные фракции является её удельная производительность при соблюдении требований точности разделения на фракции и уровня повреждений, наносимых рабочими органами.

Оптимизация взаимосвязи этих показателей является актуальным и одним из основных в проектировании калибрующих поверхностей сортировальных машин [1, 2, 5].



Основная часть

Производительность калибрующей поверхности имеет прямо пропорциональную связь со скоростью перемещения разделяемого продукта. Рассматривая калибрующие поверхности машин для сортировки картофеля, образованные продольно расположенными роликами необходимо отметить, что скорость движения материала зависит от формы клубней. Для стабилизации скорости движения клубней по поверхности используется винтовая навивка, способствующая перемещению медленно движущихся плоских клубней и ограничивающая скорость округлых, для предотвращения защемления клубней навивками соседних роликов поверхность имеет чередующиеся спиральные гладкие ролики [1, 2, 5].

Рассмотрим условие, при котором клубень, представленный в виде шара, удерживаемый выступом спирали будет находиться в равновесии. При этом на клубень будут действовать: силы трения F_1 и F_2 клубня о ролики, сила тяжести mg , силы нормальных реакций от роликов N_1 , N_2 и реакция выступа спирали N_v (рисунок 1).

Для анализа рассмотрим взаимосвязь углов γ' , θ' , ε' , ξ' , μ' , δ' , под которыми действуют силы нормальных реакций посредством которых определены проекции сил на плоскости (рисунок 2).

Центр клубня представляет собой точку O'' точки B'' , A'' и C'' места контакта клубня с роликами. При рассмотрении взаимосвязи углов и сторон.

$$O''B'' = O''C'' = R_k. \quad (1)$$

$$O''F'' = R_k - h. \quad (2)$$

$$\operatorname{tg} \gamma' = \operatorname{tg} \theta' \cdot \operatorname{tg} \varepsilon'. \quad (3)$$

$$\cos \zeta' = \frac{R_k - h}{R_k}. \quad (5)$$

$$\gamma' = \arcsin \frac{C/2 + R_1}{R_k + R_1} \quad (6)$$

где C – ширина калибрующего зазора, м; R_k – радиус клубня, м; R_1 – радиус первого ролика, м; h – высота выступа спирали, м.

Основное препятствие к движению клубня создает выступ спирали, а угол наклона калибрующей поверхности способствует росту сил стремящихся его преодолеть. Рассмотрим систему равновесия клубня на роликах

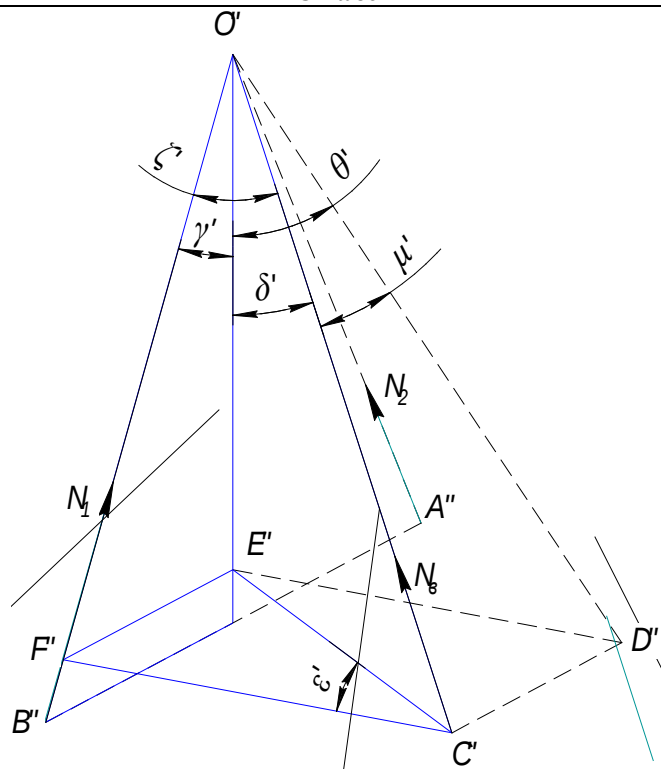


Рисунок 2 – Расчётная схема взаимодействия клубня с роликом
Fig. 2. The calculation scheme of interaction: tuber - roller

$$\begin{cases} m\ddot{x} = F_1 + F_2 - mg \cdot \sin \alpha + N_v \cdot \cos \mu' \cdot \sin \theta', \\ m\ddot{y} = (N_1 + N_2) \cos \gamma' - mg \cdot \cos \alpha + N_v \cdot \cos \mu' \cdot \cos \theta', \\ m\ddot{z} = (N_1 - N_2) \sin \gamma' + N_v \cdot \sin \gamma' \cdot \cos \zeta'. \end{cases} \quad (7)$$

Из условия(6) определена сила нормальной реакции N_2

$$N_2 = \frac{mg \cdot \cos \alpha}{2 \cdot \cos \gamma'} . \quad (8)$$

Сила реакции N_1 определена по выражению

$$N_1 = mg \left(\frac{R_k (\cos \alpha \cdot \cos \mu' \cdot \sin \theta' - \sin \alpha \cdot \cos \mu' \cdot \cos \theta')}{R_k \cos \gamma' \cdot \cos \mu' \cdot \sin \theta' - f \cdot h \cdot \cos \gamma'} - \frac{\cos \alpha}{2 \cos \gamma'} \right) .$$

Нормальная реакция на выступ спирали

$$N_s = \frac{mg \cdot \cos \alpha}{2 \cdot \cos \gamma' \cdot \cos \zeta'} - \frac{N_1}{\cos \zeta'} . \quad (10)$$

Из полученных уравнений сил нормальных реакций и конструктивных размеров роликовой калибрующей поверхности определена зависимость угла наклона калибрующей поверхности от высоты выступа спирали, радиусов роликов, размера клубня и коэффициента трения.

Получено следующее условие перехода клубня через выступ при смещении по оси $0 - z$

$$\operatorname{tg} \alpha = \frac{\operatorname{tg} \gamma (R_k \sin \delta' + f(2R_k - h))}{2 \cdot R_k \sin \delta'} . \quad (11)$$

При переходе клубня через выступ в плоскости $x - y$ за счет смещения по оси $0 - y$ и том же условии $N_1 < 0$ получена следующая зависимость $\operatorname{tg} \alpha \geq \operatorname{tg} \theta$.

Из расчетной схемы углов (рисунок 2) определен $\operatorname{tg} \theta$ как

$$\operatorname{tg} \theta = \frac{\sqrt{R_k^2 - (R_k - h)^2} \cdot (R_1 + R_k)}{(R_k - h) \cdot \sqrt{(R_1 + R_k)^2 - (R_1 + C/2)^2}} . \quad (12)$$

На основании условий полученных зависимостей (11) и (12) мы можем определить оптимальные конструктивные размеры роликов в зависимости от размеров клубней которые должны быть остановлены выступами спиралей роликов.

В процессе послеуборочной доработки картофеля основное влияние на качество закладываемого на хранение урожая оказывает операция сортирования. При сортировании удаляются растительные остатки и почвенные загрязнения, происходит разделение клубней картофеля на размерные фракции.

Показателями качества процесса калибрования являются: точность разделения, повреждения наносимые клубням рабочими органами, производительность и энергоёмкость машины в целом [3, 4].

Рассматривая роликовые калибрующие поверхности с продольным расположением рабочих органов необходимо отметить высокую точность разделения на фракции, минимальную повреждаемость клубней но при этом сравнительно низ-

кую производительность, связанную со сравнительно малой скоростью движения клубней во впадинах между роликами при рациональных углах наклона поверхности [3, 4].

Рассмотрим движение клубней по роликовой наклонной калибрующей поверхности. Клубень движется во впадине между вращающимися роликами, расположенными под наклоном. В результате исследования траектории движения клубня, выполненного в рамках предварительных опытов, было установлено, что точки поверхности клубня описывают спираль. шаг спирали зависит от формы клубня, угла наклона поверхности и частоты вращения роликов.

Для упрощения анализа предположим, что клубень имеет шарообразную форму, тогда траектория движения точки на его поверхности будет иметь вид, представленный на рисунке 3. Рассматривая данную траекторию движения, необходимо отметить, что она является функцией двух движений первое из которых, направлено вдоль впадины между роликами, второе перпендикулярно оси вращения роликов.

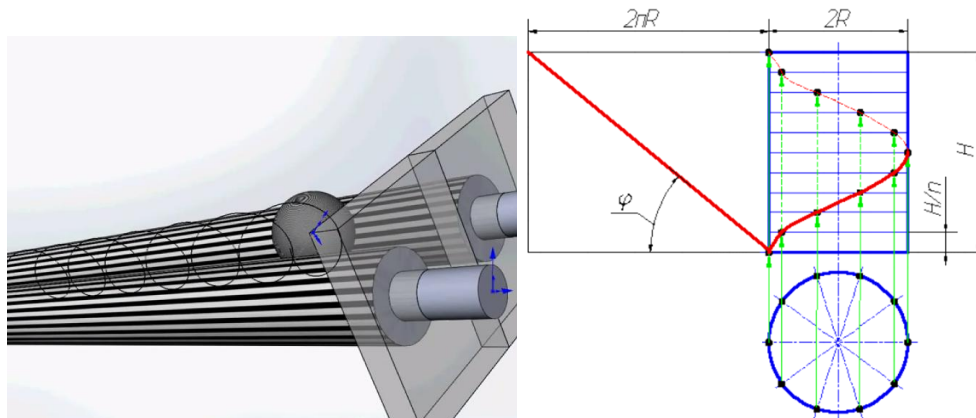


Рисунок 3 – Винтовая траектория, описываемая точкой на поверхности клубня при его движении между роликами
Fig. 3. Screw line described by the course of points of a potato tuber during its movement between the rollers

Первое движение осуществляется под действием силы тяжести, заставляющей клубни скатываться в низ. Второе за

счет вращения роликов, при условии, что ролики вращаются в одном направлении. Рассматривать данный тип движения можно как результирующую двух скоростей.

Для нахождения параметров описываемой траектории точки, рассмотрим более подробно саму винтовую линию рисунок 3. В развертке один виток будет представлять прямую линию, которая характеризуется углом подъема винтовой ψ , длиной окружности и шагом спирали H .

Таким образом, скорость движения точки может быть определена из характера траектории её движения.

Скорость клубня в векторной форме будет иметь вид

$$V_{кл} = \sqrt{V_{вращкл}^2 + V_{кач}^2}, \quad (13)$$

где $V_{вращкл}$ – составляющая от окружной скорости роликов, м/с;

$V_{кач}$ – составляющая от поступательной скорости, м/с.

Угол наклона образующей спирали траектории движения ψ определим через тангенс угла наклона.

$$\tan \psi = V_{кач} / V_{вращкл}. \quad (14)$$

Рассмотрим зависимость угловой частоты вращения клубня $\omega_{кл}$ от частоты вращения ролика $\omega_{в}$:

$$V_{вращкл} = \omega_{кл} \cdot r_{к} = \omega_{в} \cdot R_{в} = n \cdot \pi \cdot R_{в} / 30, \quad (15)$$

где $R_{в}$ – радиус ролика, м;

$r_{к}$ – усредненный радиус клубня картофеля, м;

На клубень, находящийся на наклонной поверхности во впадине между роликами, действует сила тяжести и трения, суммарная сила обеспечивающая движение клубня в сторону схода может быть определена как

$$F = m \cdot g \cdot \sin \alpha - f \cdot m \cdot g \cdot \cos \alpha, \quad (16)$$

где m – масса клубня, кг;

α – угол установки поверхности, градусы;

f – коэффициент трения клубня о материал поверхности;

g – ускорение свободного падения, кг/с.

Анализ выражения (16) показывает, что приемлемая с точки зрения удельной производительности скорость движения клубней может быть достигнута при углах наклона калибрующей поверхности свыше 30 градусов, когда качество разделения резко ухудшается. Выходом является использование роликов со спиральной наливкой.

Определим характеристики спирали исходя из требуемых эксплуатационных показателей поверхности.

Рассмотрим ускорение, как

$$a = F / m. \quad (17)$$

Поскольку ускорение это разность скоростей, то имеем следующее выражение

$$F / m = (V_{конеч} - V_{нач}) / t, \quad (18)$$

где $V_{конеч}$ – конечная скорость, м/с;

$V_{нач}$ – начальная скорость, м/с;

t – время движения на контрольном участке, с.

Время изменения скорости можем определить исходя из длины контрольного участка L поверхности.

$$t = \sqrt{2 \cdot L / a}. \quad (19)$$

$$t = (V_{конеч} - V_{нач}) / a. \quad (20)$$

Приравняв формулы (19) (20) получим и преобразовав выражение

$$V_{конеч} = a \sqrt{2 \cdot L / a} + V_{нач} = \sqrt{2 \cdot a \cdot L} + V_{нач}. \quad (21)$$

Если принять скорость движения в начале $V_{нач}=0$ м/с, то уравнение (21) примет вид

$$V_{конеч} = \sqrt{2 \cdot a \cdot L}. \quad (22)$$

Подставим значение формул (16) в (22) и преобразовав, получим

$$V_{конеч} = \sqrt{2 \cdot L \cdot g (\sin \alpha - f \cdot \cos \alpha)}. \quad (23)$$

Таким образом, скорость клубня определим как

$$V_{кл} = \sqrt{(n \cdot \pi \cdot R_{в} / 30)^2 + 2 \cdot L \cdot g (\sin \alpha - f \cdot \cos \alpha)}. \quad (24)$$

Угол наклона образующей спирали траектории движения ψ определим

$$\tan \psi = 30 \sqrt{2 \cdot L \cdot g (\sin \alpha - f \cdot \cos \alpha)} / (n \cdot \pi \cdot R_{в}). \quad (25)$$

Анализируя схему развертки спирали, рисунок 3 можем определить шаг спирали

$$\tan \psi = 2 \pi \cdot R_{в} / H. \quad (26)$$

Однако необходимо учитывать возможное защемление клубней витком спирали, которое приводит к протаскиванию клубней в щели их деформацию и повреждение. Для предотвращения данного эффекта угол спирали должен быть больше либо равен углу трения клубней о материал ролика ϕ .

$$\psi > \phi. \quad (27)$$

Заключение

Рабочие углы наклона калибрующих поверхностей с продольным положением роликов, принимают до 45 ° [1]. Перекачивание клубня через выступ по оси 0 – у возможно при следующих условиях: величина угла 22 ° высота выступа менее 2 мм, и максимальной высоте 9 мм и значении угла наклона 45 °. Обход выступа клубнем по оси 0 – z возможен при высоте в 10 мм и величине наклона поверхности в 45 °. Соответственно при значении угла 45 ° выступ 9 мм позволит удерживать клубни диаметром 90 мм.

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СОВЕРШЕНСТВОВАНИЕ КОНСТРУКЦИИ МАЛОГАБАРИТНОЙ МОЛОТКОВОЙ ДРОБИЛКИ ЗЕРНОВЫХ МАТЕРИАЛОВ

IMPROVEMENT OF CONSTRUCTION OF SMALL-SIZED HAMMER CRUSHER OF GRAIN MATERIALS

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Abstract: *For the intensification of the process of grinding grain materials, it is proposed fixed plates in the form of a ring sector additionally install on the inner surface of the hammer crusher, so that a small clearance will be provided between the lateral surfaces of the hammers and the fixed plates, in which an additional rubbing process of grinding grain material will occur.*

KEYWORDS: HAMMER CRUSHER, DESIGN, GRINDING PROCESS, INTENSIFICATION, RUBBING PROCESS

1. Введение

Молотковые дробилки используются преимущественно для среднего и мелкого измельчения сухих зерновых материалов (степень измельчения соответственно 6...10 и 10...50). По сравнению с другими типами молотковые дробилки характеризуются высокой производительностью и малым удельным расходом электроэнергии [1; 2]. С другой стороны их работу сопровождает высокий уровень шума и пылеобразования. Молотковые дробилки применяют для измельчения зерновых материалов влажностью не выше 15%.

Известные конструкции молотковых дробилок состоят из корпуса с загрузочной и выгрузочной горловинами, решета, через которое измельченный материал выводится из рабочей камеры, горизонтального ротора с шарнирно закрепленными на нем ударными элементами – молотками, и привода [3; 4].

В молотковых дробилках измельчение происходит вследствие удара молотков, которые шарнирно подвешены на осях ротора, вращающегося в рабочей камере дробилки со скоростью 500...2500 об./мин [5]. Дополнительное разрушения зернового материала происходит вследствие удара зерен о внутреннюю поверхность корпуса дробилки. Измельченный до определенной степени материал выводится из рабочей камеры измельчения через отверстия решета.

Целью модернизации малогабаритной молотковой дробилки является повышение ее производительности.

2. Предпосылки и средства для решения проблемы

Кроме процессов ударного разрушения для измельчения зерна также применяют машины, которые используют процессы раздавливания и растирания, в частности вальцевые и жерновые дробилки (мельницы).

Вальцевые дробилки раздавливают и растирают измельчаемый материал, который попал между двумя цилиндрическими вальцами, вращающихся навстречу друг другу с одинаковыми или разными скоростями (50...180 об./мин). Вальцевые дробилки с гладкими вальцами обеспечивают степень измельчения 3...4, а с зубчатыми вальцами – 4...6.

Жерновые мельницы состоят из неподвижного и подвижного камней (жернова). Подвижный камень вращается со скоростью 20...50 об./мин. Жерновые мельницы применяют для мелкого и тонкого измельчения зерновых материалов (соответственно степень измельчения 10...50 и 50...100). Вследствие недостаточной производительности жерновые мельницы редко применяют для измельчения зерна.

Валковые дробилки и жерновые мельницы характеризуются невысокой производительностью и сравнительно высокими энергозатратами.

Преимуществом валковых и жерновых дробилок является возможность измельчения влажных зерновых материалов.

3. Решение рассматриваемой проблемы

Поставленная задача повышения производительности малогабаритной молотковой дробилки решается тем, что одновременно с процессом ударного разрушения предлагается использовать и процесс перетирания зернового материала.

4. Результаты и дискуссия

Для обеспечения комбинированного процесса разрушения зернового материала вследствие удара и перетирания в малогабаритной молотковой дробилке на внутренней поверхности корпуса предлагается дополнительно установить неподвижные пластины в виде сектора кольца [6]. Установление таких пластин способствует интенсификации процесса измельчения за счет дополнительного перетирания измельчаемого материала, который попадает в пространство между движущимся молотом и неподвижной пластиной.

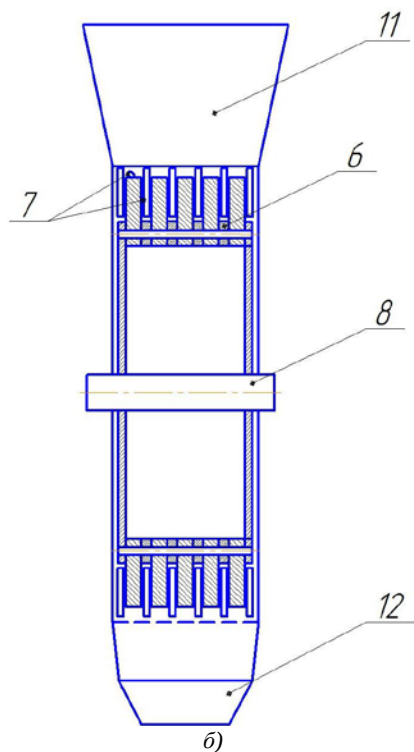
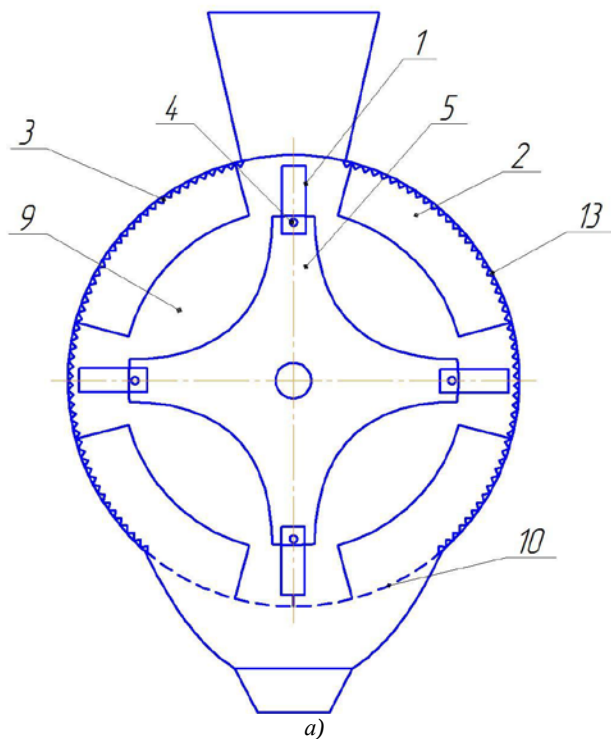
На фиг. 1, а схематически изображена малогабаритная молотковая дробилка комбинированного действия с установленными неподвижными пластинами в виде сектора кольца и его поперечный разрез (фиг. 1, б).

Малогабаритная молотковая дробилка комбинированного действия содержит загрузочную 11 и выгрузочную 12 горловины, корпус 3 с рифлёной внутренней поверхностью 13, в середине которого находится рабочая камера 9 с ротором. Ротор состоит из вала 8, на котором установлены боковые диски 5. К этим дискам 5 прикреплены оси 4, на которых шарнирно подвешены молотки 1. В нижней части корпуса 3 установлено решето 10, через которое осуществляется вывод продуктов измельчения. На внутренней поверхности корпуса неподвижно закреплен набор пластин таким образом, что каждая пластина 2 расположена на одинаковом расстоянии между двумя соседними молотками 1.

Схема взаимодействия частицы измельчаемого материала с боковыми поверхностями молотка 1 и пластины 2 малогабаритной молотковой дробилки комбинированного в процессе разрушения путем перетирания изображена на фиг. 2.

Дробилка работает следующим образом.

Зерновой материал попадает в рабочую камеру через загрузочную горловину 11. Во время движения вала 8 вращаются диски 5 с молотками 1, которые шарнирно закреплены на осях 4. В результате удара молотком 1 зерно 7 частично разрушается и отбрасывается к рифлёной поверхности 13 корпуса 3. Вследствие удара об эту рифлёную поверхность продолжается процесс разрушения измельчаемого материала. Кроме того, зерно разрушается, ударяясь об торец пластины 2.

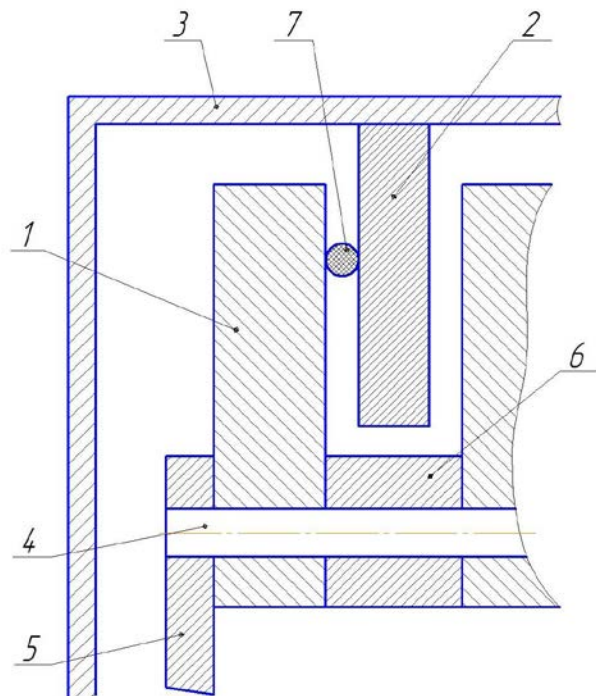


Фиг. 1. Схема малогабаритной молотковой дробилки комбинированного действия (а) и ее поперечный разрез (б):

1 – молоток, 2 – неподвижные пластины, 3 – корпус, 4 – ось, 5 – диск, 6 – втулка, 7 – измельчаемого материала, 8 – вал, 9 – рабочая камера, 10 – решето, 11 – загрузочная горловина, 12 – выгрузной горловина, 13 – рифлёная внутренняя поверхность корпуса.

Дополнительно, благодаря установленным недвижимым пластинам, происходит процесс перетирания измельчаемого материала, который попал в пространство между молотком 1 и пластиной 2, при относительном движении молотка 1 вдоль неподвижной пластины 2.

Равномерное размещение молотков 1 относительно неподвижных пластин 2 обеспечивается с помощью втулок 12.



Фиг. 2. Взаимодействие частицы измельчаемого материала с боковыми поверхностями молотка и пластины малогабаритной молотковой дробилки комбинированного действия:

1 – молоток, 2 – неподвижные пластины, 3 – корпус, 4 – ось, 5 – диск, 6 – втулка, 7 – измельчаемого материала.

Измельчение осуществляется также в результате взаимодействия между зернами и продуктами измельчения.

Продукты измельчения просеиваются через сито 10 и удаляются из машины через выгрузную горловину 12.

5. Заключение

Усовершенствованная конструкция малогабаритной молотковой дробилки будет способствовать интенсификации процесса измельчения зерновых материалов за счет одновременной реализации процессов их перетирания и ударного разрушения.

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ОПРЕДЕЛЯНЕ СТЕПЕНТА НА ВЛИЯНИЕ НА ОСНОВНИТЕ СТРУКТУРНИ ПАРАМЕТРИ ВЪРХУ ТЕХНИКО-ИКОНОМИЧЕСКИТЕ ХАРАКТЕРИСТИКИ НА ДВГ

DETERMINATION OF THE DEGREE OF INFLUENCE OF MAIN STRUCTURAL PARAMETERS ON THE CHANGE OF INTERNAL COMBUSTION ENGINE TECHNICAL AND ECONOMIC CHARACTERISTICS

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Abstract: The evaluation of the hourly and specific fuel consumption and the main components of the exhaust gas composition in case of internal combustion engine failures are examined. A structural investigation scheme was developed to determine the impact of the structural parameters of the main subsystems on the change of the main components of the exhaust gases.

Keywords: DIAGNOSTIC, FUEL CONSUMPTION, COMPOSITIONS EXHAUST GAS, INTERNAL COMBUSTION ENGINE

1. Въведение

Известно е, че в съвременните дизеловите ДВГ без бордова електронна система (БЕС), се състоят от голям брой отделни механични подсистеми, които не се различават съществено в конструктивно отношение. Поради голямото разнообразие от работни повърхнини, характеризиращи се със съответните структурни параметри е възможно възникването и на различни повреди. За тяхното установяване и локализиране е необходимо да се използват както интегрални, така и частни диагностични параметри с различни свойства. Между тях и съответстващите им структурни параметри съществуват сложни функционални и стохастични връзки. Ето защо за опростяване на задачата при разработването на диагностичните таблици и тестове, и алгоритми и е необходимо условно да бъдат разделени на отделни подсистеми, при които в голяма степен съществува функционална независимост.

Известно е, [1, 3, 4, 5] че в зависимост от динамиката на изменение на структурните параметри при отделните работни повърхнини и тяхното съчетание, същите оказват съществено влияние на диагностични параметри въз основа, на които се определя техническото състояние на обекта. За определяне на степента на влияние на отделните повреди и на тяхното едновременно възникване е необходимо да се определят и критерии, чрез които да се определи влиянието им върху разхода на гориво и върху съотношението на основните компоненти в отработените газове. За определянето на тези коефициенти е необходимо да се познават конструктивните особености на съответните подсистеми, които оказват съществено влияние върху динамиката на изменение на контролираните диагностични показатели.

2. Изложение

В редица литературни източници [1, 3] са посочени различни методики за разработването на оптимални или рационални алгоритми, без да са взети в предвид различните възможни съчетания на отделните повреди, свързани с реда на извършване на елементарните проверки. Това дава възможност да се минимизират разходите, свързани с поставянето на съответната диагноза.

Като изходни данни за определяне на аналитичната връзка между диагностичните и структурните параметри, при различно съчетаване на отделните повреди, са използвани структурно-следствените схеми, приведени в [2].

Въз основа на анализа на тези резултати е съставена следната система уравнения:

$$U_1 = f(X_1, X_2, X_{15}, X_{16}) \quad (1)$$

$$U_2 = f(X_3, X_4, X_5, X_6, X_7, X_8, X_9, X_{10}, X_{11}) \quad (2)$$

$$U_3 = f(X_2, X_4, X_7, X_8, X_9, X_{10}, X_{11}, X_{13}, X_{14}, X_{18}) \quad (3)$$

$$U_4 = f(X_5, X_6, X_8, X_9, X_{10}, X_{12}, X_{13}, X_{14}) \quad (4)$$

$$U_5 = f(X_{11}, X_{15}, X_{16}, X_{17}) \quad (5)$$

Горната система от пет уравнения описва всички възможни състояния на обекта, подлежащ на диагностиране и дава функционалната връзка между структурните и частните диагностични параметри, които влияят върху динамиката им на изменение. Въз основа на анализа на определените коефициенти на структурна информативност и коефициентите на неразличимост (табл.1) се вижда, че последните се изменят в границите от 0,1 до 0,25 и от 0,33 до 1,0. За да се обоснове окончателно броя от диагностични параметри и определи реда за тяхното измерване е необходимо да се изследва връзката между структурните параметри X и диагностичните параметри U . Поради стохастичния характер на изходящите процеси и грешките при измерване на диагностичните параметри, в общия случай може да се говори за статистическа връзка между диагностичния и структурния параметър в диапазона от номиналната X_n до граничната му стойност $X_{гр}$. Следователно по отношение на математическото очакване на параметрите са в сила зависимостите: $U_j = f(X)$ или $X = \varphi(U_j)$, където $j=1,2,3,...,m$ е броят изходящи параметри, които могат да се оценят обективно и точно. От анализа на уравненията $U_j = f(X)$ влизащи в горепосочената система се вижда, че различните повреди могат да окажат влияние върху съответното съотношение на компонентите влизащи в състава на отработените газове. От анализа на посочените по-горе уравнения се вижда, че съществува определена повтораемост, относно влиянието на структурните и съответните им диагностични параметри.

Така например X_8, X_9, X_{10} оказват влияние върху три диагностичните показатели - U_2, U_3 и U_4 ; X_4 и X_7 влияят върху U_2 и U_3 ; X_{13}, X_{14} – върху U_3 и U_4 ; X_2 – върху U_1 и U_3 ; X_1 – върху U_1 и U_5 ; X_6 – върху U_2 и U_4 ; Останалите повреди оказват влияние само върху един частен диагностичен параметър. Следователно възниква въпросът в каква последователност трябва да бъдат извършени елементарните проверки, така че да се достигне най-бързо до поставяне на диагнозата при минимум разход на труд и средства. От така представената матрица е установено, че между частните диагностични параметри и основните групи структурни параметри съществува двупосочна връзка. В зависимост от тяхното съчетаване е установено, че това води до различна степен на интензивност на влошаване на технико-икономическите показатели на ДВГ и на състава на отработените газове. За определяне на рационалната последователност при извършване на елементарните проверки и установят причинно-следствените връзки за моментното състояние на контролирания обект е необходимо да се анализира системата от уравнения свързани с $X = \varphi(U_j)$. Въз основа на матрицата (табл.1) системата от уравнения, изразяващи

функционалната връзка между структурния параметър и съответстващите му диагностични. На базата на информацията от табл.1 системата уравнения може да се представи във вида:

$$\begin{aligned} X_1 &= \varphi(U_1); & X_{10} &= \varphi(U_2, U_3, U_4) \\ X_2 &= \varphi(U_1, U_5, U_7) & X_{11} &= \varphi(U_2, U_3) \\ X_3 &= \varphi(U_2) & X_{12} &= \varphi(U_4) \\ X_4 &= \varphi(U_2, U_3) & X_{13} &= \varphi(U_3, U_4) \\ X_5 &= \varphi(U_2, U_4) & X_{14} &= \varphi(U_3, U_4) \\ X_6 &= \varphi(U_2, U_4) & X_{15} &= \varphi(U_1, U_5) \\ X_7 &= \varphi(U_2, U_3) & X_{16} &= \varphi(U_1, U_5) \\ X_8 &= \varphi(U_2, U_3, U_4) & X_{17} &= \varphi(U_5) \\ X_9 &= \varphi(U_2, U_3, U_4) & X_{18} &= \varphi(U_3) \end{aligned}$$

Таблица 1.

Матрица за взаимовръзката между частните диагностични параметри и структурните параметри на хранителната система на двигатели без БЕС

№	Неизправност	Диагностичен параметър					Коефициент на неразличимост
		РВК	ЦКГ	МПГ	РПГ	КВБЦГ	
		U ₁	U ₂	U ₃	U ₄	U ₅	
1	Неизправен въздушен филтър	+	-	-	-	-	1,0
2	Топлинна хлабина	+	-	+	-	+	0,33
3	Износване на подкачващата помпа	-	+	-	-	-	1,0
4	Замърсен горивен филтър.	-	+	+	-	-	0,5
5	Износване на Б.Ц. от ГНП	-	+	-	+	-	0,5
6	Износване на преливния клапан	-	+	-	+	-	0,5
7	Износване на регулатор	-	+	+	-	-	0,5
8	Износване на разпръсквачите	-	+	+	+	-	0,33
9	Закоксуване на разпръсквачите	-	+	+	+	-	0,33
10	Еластичност и налягане на пружините на иглата	-	+	+	+	-	0,33
11	Износване на нагнетателния клапан	-	+	+	-	-	0,5
12	Неравномерност на износване на Б.Ц. от ГНП.	-	-	-	+	-	1,0
13	Характеристики на пружините на ГНП, регулатора	-	-	+	+	-	0,5
14	Износване на гърбичния вал от ГНП	-	-	+	+	-	0,5
15	Хлабина бутало-цилиндър	+	-	-	-	+	0,5
16	Еластичност на буталните пръстени	+	-	-	-	+	0,5
17	Нагар в над буталното пространство	-	-	-	-	+	1,0
18	Неправилна регулировка на двигателя	-	-	+	-	-	1,0
	Коефициент на структурна информативност	0,25 (1/4)	0,111 (1/9)	0,1 (1/10)	0,125 (1/8)	0,25 (1/4)	

Легенда: РВК – разреждане в всмукателния колектор; ЦКГ – циклово количество гориво; МПГ – момент на подаване на горивото; РПГ – равномерност на подаване на горивото; КВБЦГ – компресия и вакуум в бутало-цилиндровата група.

От анализа на показаните резултати в табл.1, и изведените аналитичните зависимости между структурните и диагностичните параметри може да се стигне до заключението, че за разработването на рационален алгоритъм тези резултати са необходими, но не и достатъчни. Това произтича от получените експериментални резултати, показани в табл.2 и табл.3 и на фиг.1 - фиг.7. От тези резултати се вижда, че върху количественото изменение на интегралните диагностични параметри съществено влияние оказват динамиката на изменение на посочените в табл.5.1 частни диагностични показатели, зависещи от съчетаването на изменението на структурните параметри. Последните оказват нееднозначно

влияние върху изменението на часовият и специфичен разход на гориво, а така също и върху съотношението на отделните компоненти в състава на отработените газове (табл.2 и табл.3). Ето защо за установяването на рационална последователност по отношение извършването на елементарните проверки е необходимо да се определи и степента на влияние на контролираните параметри посочени в табл.5.1 върху интегралните диагностични показатели. За целта допълнително са определени коефициента на чувствителност за изменение на контролираните диагностични показатели спрямо номиналните стойности на частните диагностични параметри и тяхното процентно отклонение спрямо големините на контролираните показатели съответстващи на изправен двигател (табл.2, табл.3 и от фиг.1 до фиг.7).

Таблица 2.

Изменение часовия разход на гориво и състава на отработените газове във функция от мощността на ДВГ при различни неизправности

№	Параметър	Стойност	Гч, l/h	HC, ppm	CO, %	CO ₂ , %	O ₂ , %	NO _x , ppm	λ
1	пад на налягане	20 kPa *	11.98	21	0.026	6.35	10.04	1094	1.43
		80 kPa	12.56	23	0.043	7.59	11.1	1004	2.29
		140 kPa	13.02	25	0.083	8.01	12.31	907	2.75
2	замърсеност на ВФ	0 kPa *	11.98	21	0.026	6.35	10.04	1094	1.43
		10 kPa	13.50	34	0.088	7.97	8.39	1128	1.36
		20 kPa	14.72	47	0.169	8.77	6.53	1167	1.16
3	топлинна хлабина	0,15 mm	13.01	63	0.051	6.53	11.26	790	2.09
		0,45 mm *	11.98	21	0.026	6.35	10.04	1094	1.43
		0,75 mm	14.27	26	0.07	6.75	10.58	859	1.89
4	налягане на сработване на дюзата	15 MPa	12.58	51	0.033	6.56	10.58	865	2
		17,5 MPa *	11.98	21	0.026	6.35	10.04	1094	1.43
		20 MPa	13.45	66	0.039	6.74	11.18	830	2.21
5	ъгъл на впръскване	14°	14.76	39	0.183	7.19	10.72	232	1.72
		23° *	11.98	21	0.026	6.35	10.04	1094	1.43
		31°	13.20	29	0.167	6.64	11.13	1199	2.11

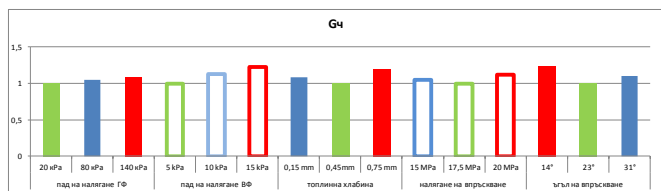
* стойности на параметрите при изправен ДВГ

Таблица 3.

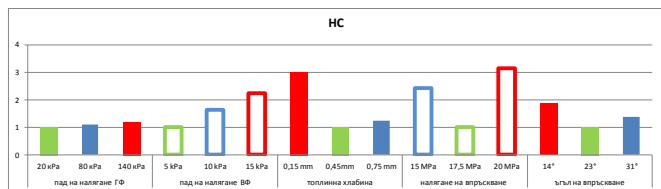
Коефициент на чувствителност за изменение на контролираните диагностични показатели спрямо номиналните стойности на структурните параметри

№	Параметър	Стойност	Гч, l/h	HC, ppm	CO, %	CO ₂ , %	O ₂ , %	NO _x , ppm	λ
1	пад на налягане ГФ	20 kPa *	1	1	1	1	1	1	1
		80 kPa	1.048	1.095	1.654	1.195	1.11	0.91	1.60
		140 kPa	1.087	1.190	3.192	1.261	1.23	0.83	1.92
2	пад на разреждане ВФ	0 kPa *	1	1	1	1	1	1	1
		10 kPa	1.126	1.619	3.385	1.255	0.84	1.03	0.95
		20 kPa	1.229	2.238	6.5	1.381	0.65	1.07	0.81
3	топлинна хлабина	0,15 mm	1.086	3.000	1.962	1.028	1.12	0.72	1.46
		0,45mm *	1	1	1	1	1	1	1
		0,75 mm	1.191	1.238	2.692	1.063	1.05	0.79	1.32
4	налягане на сработване на дюзата	15 MPa	1.050	2.429	1.269	1.033	1.06	0.79	1.40
		17,5 MPa *	1	1	1	1	1	1	1
		20 MPa	1.123	3.143	1.500	1.061	1.11	0.76	1.54
5	ъгъл на впръскване	14°	1.232	1.857	7.038	1.132	1.07	0.21	1.20
		23° *	1	1	1	1	1	1	1
		31°	1.101	1.381	6.423	1.046	1.11	1.09	1.48

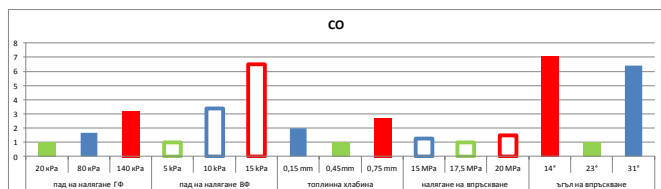
* стойности на параметрите при изправен ДВГ



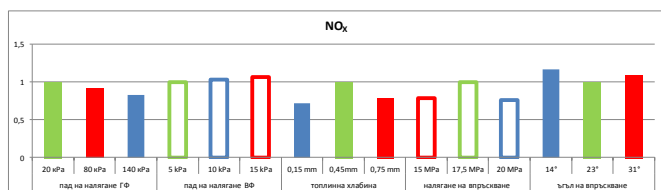
Фиг. 1. Изменение коэффициента на чувствителност за часовия разход на гориво спрямо номиналната стойност при изправен двигател



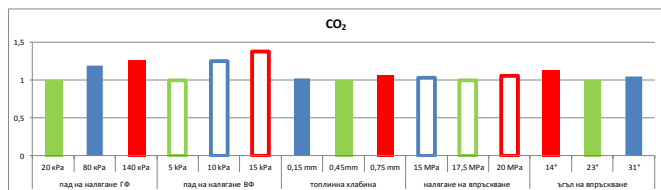
Фиг. 2. Изменение коэффициента на чувствителност за количеството на HC спрямо номиналната стойност при изправен двигател



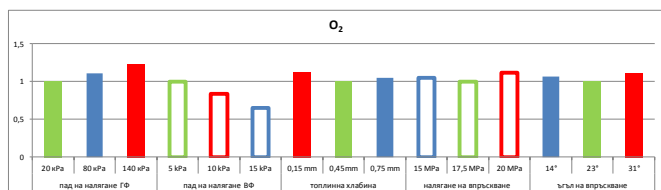
Фиг. 3. Изменение коэффициента на чувствителност за количеството на CO спрямо номиналната стойност при изправен двигател



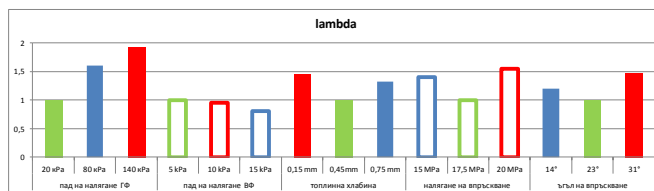
Фиг. 4. Изменение коэффициента на чувствителност за количеството на NOx спрямо номиналната стойност при изправен двигател



Фиг. 5. Изменение коэффициента на чувствителност за количеството на CO₂ спрямо номиналната стойност при изправен двигател



Фиг. 6 Изменение коэффициента на чувствителност за количеството на O₂ спрямо номиналната стойност при изправен двигател



Фиг. 7. Изменение коэффициента на чувствителност за λ спрямо номиналната му стойност при изправен двигател

За определяне реда при извършване на елементарните проверки от съществено значение е да се вземат предвид онези фактори, които оказват най-голямо негативно влияние, както върху експлоатационните разходи свързани основно с разхода на гориво смазочните материали, така също и влиянието им върху околната среда. Въз основа на показаните таблични и графични зависимости се вижда, че тези фактори условно могат да бъдат разделени на три групи: 1. Степен на влияние на съответните структурни параметри върху часовия разход на гориво; 2. Степен на влияние на структурните параметри върху токсичните и канцерогенни компоненти в състава на ОГ; 3. Степен на влияние на структурните параметри върху не токсичните компоненти. От съществено значение при разработването на алгоритмите е да се вземат предвид първите две групи. От анализа на приведените по-горе резултати се вижда, че при определени случаи съществува еднопосочност на изменение на контролираните параметри. Освен това процентното отклонение на часовия разход на гориво се движи в границите от 4.30 до 23.20 % при регламентирани гранични отклонения от 3 до 4 %. При такова увеличаване разхода на гориво най-често води до работа на двигателя при обогатена горивна смес. В областите с обогатена смес емисията на HC нараства поради невъзможността цялото количество HC да изгорят от една страна, а от друга в резултат на недостатъчното количество O₂. Върху отделянето на HC влияят още правилното функциониране и регулировка на запалителната система и състоянието на механичната част на двигателя. Не херметичността в смукателната система, не херметичността в бутало цилиндровата група, наличието на твърди отложения по фаските на клапаните са причина за голямо количество не изгорели HC в отработените газове. Друга причина е увеличаване разхода на масло. Увеличеното количество HC заедно с наличието на CO е съществена предпоставка за провеждане на задълбочена диагностика. Ето защо това ни дава основание да извършим онези проверки за U₁ и U₃ посочени в табл.1 и уравнения 1 и 3, като последователността се определи въз основа на последователните процедури с прилагането на условните диагностични програми. Едновременно с това при ранжирането на съответните проверки е необходимо да се има предвид контролопригодността на съответната подсистема и наличието на неразличимост по отношение на съответните повреди. Подобни закономерности се получават и при останалите резултати получени в процеса на експериментите. Тъй като определяне състава на отработените газове може да се отнесе към експресните диагностични методи без да има определени изисквания по отношение на контролопригодността на контролирания обект, то при определяне реда на извършване на съответните проверки, началния етап при разработването на алгоритмите е целесъобразно да става въз основа на интензивността на изменени на токсичните компоненти. Окончателното ранжиране на извършване на елементарните проверки с установяване на причинно следствените връзки е целесъобразно да се извърши въз основа на данните от табл.1 и системата уравнения за функционалната връзка между диагностичните и структурни параметри.

3. Заключение

1. Въз основа на анализа за изменението на коефициента за структурна информативност е установено, че за интегрални диагностични показатели е целесъобразно да се използват часовия разход на гориво и състава на отработените газове. Предимството на втория интегрален показател се изразява в незначителните разходи за труд и време необходими за тяхното определяне в производствени условия и без особени изисквания по отношение за контролопригодност на съответния обект.

2. Определени са основните структурни параметри и тяхното съчетание на съответните подсистеми, свързани с интензивността на изменение на основните диагностични показатели и на тази база са определени коефициентите на структурна информативност и неразличимост.

3. Въз основа на получените резултати са определени критерии за определяне реда на извършване на елементарните проверки и, които служат за база при разработването на рационален алгоритъм за диагностика на хранителната система на ДВГ без БЕС.

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РАЗРАБОТВАНЕ НА АЛГОРИТЪМ ЗА ДИАГНОСТИКА НА ДВГ, ПРИ ИЗВЕСТЕН КОЕФИЦИЕНТ НА СТРУКТУРНА ИНФОРМАТИВНОСТ

DEVELOPING ALGORITHM FOR DIAGNOSIS OF INTERNAL COMBUSTION ENGINE WITH KNOWN COEFFICIENT OF STRUCTURAL INFORMATION

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Abstract: On the basis of the results obtained for the influence of a major engine failure on the exhaust gas composition, criteria have been set as a basis for the development of a rational algorithm related to the order of execution of the elementary checks. For this purpose and a coefficient of structural information, the degree of influence on fuel consumption and the ratio of the major components in the exhaust gases for one or more faults or their respective combination are determined.

Keywords: DIAGNOSTIC, FUEL CONSUMPTION, COMPOSITIONS EXHAUST GAS, INTERNAL COMBUSTION ENGINE

Въведение

Известно е, че при алгоритмите за диагностика проверките се извършват в определен ред и резултатът от една проверка носи известна информация за състоянието на обекта на диагностика. Ако след всяка проверка се анализира получаваната информация, може повредата да се локализира, преди да се реализира съответния алгоритъм и извършването на останалите проверки следва да се прекрати. Алгоритъмът за диагностиране в най-общия случай се състои от определяне съвкупността от т.н. елементарни проверки на обекта, а така също от правила, установяващи последователността от реализация на елементарните проверки и правила за анализ на резултатите от последните. Всяка елементарна проверка се определя от тестови или работни въздействия, подавани или постъпващи към обекта и състава на контролните точки, от които се получават отговорите на обекта за това въздействие. Диагнозата се поставя в общия случай по съвкупност от получените резултати на елементарните проверки [1, 5]. Следователно възниква въпроса каква трябва да бъде стратегията в търсенето на повредите и оптималната последователност на проверките.

Известно е [3, 4], че оптимизирането на алгоритмите за диагностиране е възможно тогава, когато броят на елементарните проверки е достатъчен за решаване на конкретните задачи за диагностиране и е по-малко от всички възможни елементарни проверки за даденото изделие. За различните елементарни проверки се правят различни разходи за тяхната реализация. От тези проверки се получава различна информация за техническото състояние на обекта. Освен това, едни и същи елементарни проверки могат да бъдат реализирани в различна последователност. Ето защо за решаването на едни и същи задачи при диагностирането могат да бъдат съставени няколко алгоритъма, които се различават или по съставните елементарни проверки, или в последователността на тяхната реализация или пък по необходимите разходи за тяхната реализация.

Изложение

За разработването на алгоритъм за диагностиране на определена подсистема от ДВГ без БЕС е в зависимост не само от контролопригодността на обекта за диагностика, но е необходимо да се вземе предвид коефициентите за структурна информативност и чувствителност за изменение на диагностичните параметри.

За целта е необходимо да се установи влиянието на горепосочените фактори върху изходните параметри и определи рационалното им съчетание така, че да се минимизират разходите свързани с поставянето на диагнозата. Въз основа на получените експериментални резултати показани в [дис Бари, Евгени], а така също от анализа на графичните и аналитични зависимости се вижда, че съществува сложна взаимовръзка между всички тези параметри. Съгласно златното правило при диагностиката изразяващо се в: от по-голямата вероятност към

по-малката, от простото към сложното е необходимо при разработването на алгоритми да се спазват следните принципи:

1. Да се започне с онези елементарни проверки, които се намират във функционална връзка със съответните свойства на контролираните параметри, при които коефициента за структурна информативност е най-голям. Това произтича от факта, че колкото големината на този коефициент е по-малка, то върху него ще оказват влияние по-голям брой структурни параметри, следователно и броят на елементарните проверки ще бъде по-голям поради което времето и разхода на труд ще са по-големи.

2. Да се има предвид коефициента за неразличимост. Трябва да се започне от онези елементарни проверки, при които коефициента на неразличимост е най-голям по абсолютна стойност. Това означава, че при коефициент на неразличимост равен на единица ще получим информация за съответния диагностичен параметър, само от една елементарна проверка. Горното твърдение може да бъде в сила при условие, че дадения диагностичен параметър зависи само от един структурен параметър. От анализа на уравненията (5.1 до 5.5) посочени в [2] се вижда, че съществува много сложна взаимовръзка между структурните и диагностични параметри, което затруднява твърде много решаването на поставената задача. Следователно горното твърдение може да се приеме като необходимо, но не и достатъчно условие. За решаването на този казус е необходимо да се вземат предвид и други ограничителни условия които да служат като спомагателни фактори при вземането на едно или друго решение относно последователността при извършване на съответните елементарни проверки. За целта е необходимо:

- да се извърши анализ на частните диагностични параметри относно интензивността им на изменение и определи степента на влияние на отделните групи фактори върху съотношението на отделните компоненти влизащи в състава на ОГ и степента им на влияние върху обобщените диагностични параметри. Въз основа на резултатите от анализа същите да се разположат в монотонно намаляваща редица при което се получава ред, който характеризира степента на влияние на тези групи фактори върху съответните свойства, върху интегралния показател.

$$U_1 > U_2 > U_3 \geq \dots U_i \quad (1)$$

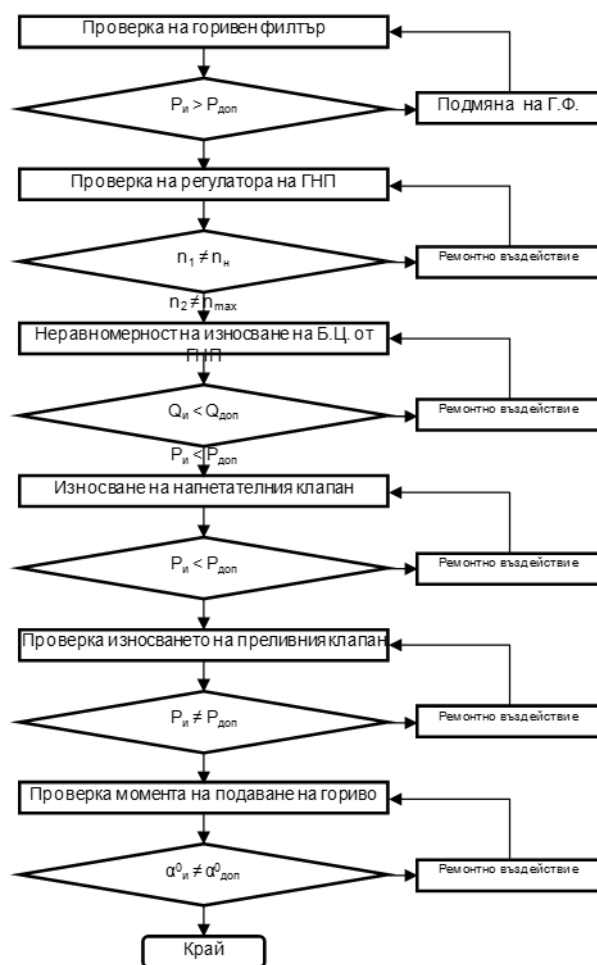
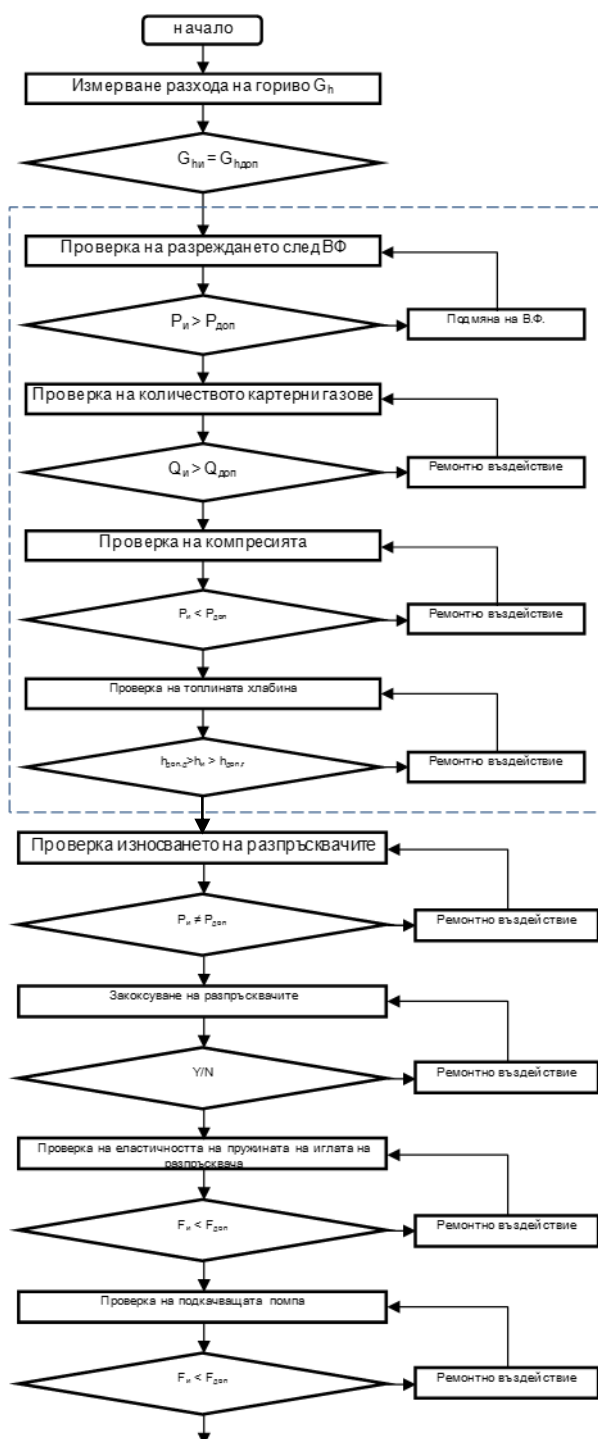
- след определяне на съответния диагностичен параметър с най-голямо влияние върху интензивността на изменение на съответните свойства е необходимо да се извърши анализ на всички структурни параметри, които оказват влияние върху него. Необходимо е да се започне от онези диагностични параметри, чиито числена стойност на коефициента на неразличимост е най-голяма. Това означава, че броят на елементарните проверки за определяне на съответния структурен параметър е минимален. Горното твърдение произтича от факта, че ако контролирания диагностичен

параметър се намира в границите на допустимите стойности, следователно отпада необходимостта за извършване на тази елементарна проверка.

- от множеството контролирани съставки на ОГ е необходимо да се започне с контролирането на онези компоненти, които имат най-голямо отклонение от допустимите стойности и притежават определени токсични свойства, каквито в случая са НС и СО. При условие, че изменението на обобщения диагностичен параметър във функция от големината на структурните параметри представлява монотонно нарастваща функция са възможни три варианта на диагнозата: $U_{об} > U_{гр}$; $U_{доп} < U_{об} < U_{гр}$; $U_{об} < U_{доп}$. Ако изменението на контролираните показатели се променя по монотонно намаляваща функция, то възможните три варианта ще имат аналогичен характер изразяващи се с неравенствата: $U_{об} < U_{гр}$; $U_{доп} > U_{об} > U_{гр}$; $U_{об} > U_{доп}$. При първите варианти не е необходимо контролиране на останалите показатели, тъй като $U_{об}$ е по-голямо, респективно по-малко от граничните стойности. При втория вариант е необходимо контролиране и

на останалите показатели и при условие, че са със стойности по-малки от $U_{гр}$ и е необходимо да се определи остатъчния ресурс спрямо онзи показател, чиито стойности са най-близки до граничните. При третия вариант не е необходим допълнителен контрол до следващата междудиагностична проверка. При условие, че е налице вторият вариант, то контролирането на съответните диагностични показатели е необходимо да се извършва по реда на ранжирането им в една монотонно намаляваща функция съобразно тяхната интензивност на изменение. Извършването на контрола в съответния рационални алгоритми е необходимо да става в ред съобразен с горе посочените изисквания в резултат на което се осигурява пълното използване ресурса на контролираният обект при минимални експлоатационни разходи [2]

От получените резултати приведени в [2] е установено, че една от причините за наличие на голямо количество СО, НС и повишен разход на гориво е свързана със степента на разреждането в смукателния колектор. Ето защо проверките е необходимо да се извършва в ред: състояние на въздушния



Фиг. 1. Алгоритъм за диагностика.

филтър X_1 при който Кси е най голям и е равен на 0.25, а коефициента на неразличимост е 1.0. Следващата проверка е свързана с определяне на компресията и вакуума в БЦГ изразени със X_{15} и X_{16} при които коефициента на неразличимост е 0.5. Последната проверка за U_1 е свързана с определяне на топлинната хлабина X_2 с коефициент на неразличимост 0.25.

Тъй като при отклонение от ъгълът на впръскване имаме най-голямо отклонение от нормативните показатели по отношение разхода на гориво и наличието на токсични газове, то следващата проверка е свързана с елементарните проверки свързани с U_3 . На базата на тези разсъждения са определени и реда за извършване на останалите проверки, чиито последователност се изразява в: $U_3 - X_8, X_9, X_{10}; X_3; X_4, X_7; X_5; X_{11}; X_6; X_{18}$. За определянето на U_2 от възможните девет проверки е необходимо да се извършат само три: за $X_3; X_5; X_6$, а за U_4 само една – X_{12} . От така извършеното ранжиране на факторите и след като са взети гореизброените условия е разработен и съответният алгоритъм, показан на фиг.1.

От горепосоченият анализ се вижда, че при прилагане на съответната методика и построяването на алгоритъма свързан с реализирането на пълна диагностика за разглежданите подсистеми от възможният общ брой 36 елементарни проверки, същите могат да бъдат редуцирани до 18. Определянето реда за провеждане на горните проверки е извършен въз основа на изискването за големината на коефициента на неразличимост, коефициента за структурна информативност и отклонението на токсичните компоненти от нормативните изисквания. Както беше отбелязано по-горе първата проверка се започва от онази група частни диагностични показатели, респективно структурен параметър, при които коефициента на неразличимост е най-голям т.е. за определяне на неговата големина е необходимо да се извършат минимален брой проверки. При условие, че един или два от контролираните показатели са с по-голяма интензивност на изменение спрямо останалите, това означава, че е необходимо да се открие причинно следствената връзка за протичането на тези процеси. За целта е необходимо да се назначи провеждането на задълбочена дискретна диагностика, като се започне от онези възли при които състоянието на съответните контролирани показатели зависи от минимален брой структурни параметри, т.е. необходимо е провеждането на минимум елементарни диагностични проверки. За целта е необходимо да се разработят структурно-следствените или функционални схеми и се започне от онези възли, чиито структурна информативност е най-голяма. Въз основа на тези схеми е разработен рационален алгоритъм, като за целта последователността на извършване на елементарните проверки е съобразена със съответната стратегия за търсене на неизправностите и оптималната последователност на проверките. При наличието на много функционална връзка, както е нашия случай е използван принципа на последователните процедури, които се извършват в определен ред по предварително разработени схеми, като всяка следваща проверка се определя в зависимост от резултатите на предходната.

Заклучение

От приведените по-горе графики за изменение количеството на основните компоненти в ОГ се вижда, че наред със съотношението между количеството въздух и горивото, най-голямо влияние върху емисията на вредните вещества в ОГ оказва момента на възпламеняването на горивната смес. При по-ранно възпламеняване количеството на НС се увеличава с 18 %, а емисията на NO_x с 12 % при ъгъл на възпламеняване от 31° . Разходът на гориво и емисията на вредните вещества се намират в обратно пропорционална връзка. С повишаване коефициента на излишък от въздух за компенсация на не голямата скорост на изгаряне е необходимо и по-ранно възпламеняване на горивната смес, за да се осигури оптимален процес на изгаряне. За осигуряването на рационален компромис в зависимост от оборотите и натоварването на двигателя, е необходима сложна регулировка на момента на запалването.

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ИЗСЛЕДВАНЕ НА ТЕХНОЛОГИИ И МАШИНИ ЗА ЗАЩИТА НА ЛОЗОВИТЕ НАСАЖДЕНИЯ

RESEARCH OF GRAPEVINE PROTECTION TECHNIQUES

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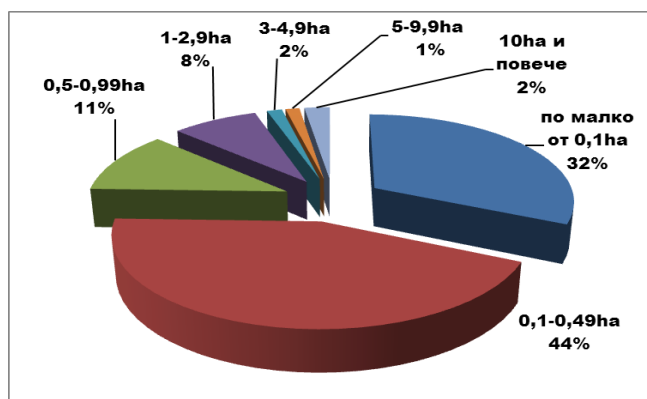
Abstract: The present research shows and analyses the most important grapevine protection techniques. The characteristics of various methods for spraying and transporting the spray material have been determined in order to obtain optimum results. Using a trap screen allows to reduce the percentage of spray solution loss by capturing part of it and reducing the risk of environmental contamination.

As a result of reflecting nozzle output, the trap screen of air blast sprayers improves their performance and increases the coverage from one row to two. The trap screen of air blast sprayers working without a fan unit does not increase the performance of the most important inner and lower parts and should not be used in spite of enhancing the performance in the upper and central parts. The use of mounted sprayers and aerial application can be limited, only in those cases when spray coverage on the reverse side of the leaf is not required.

KEYWORDS: MECHANIZATION, MACHINERY FOR PLANT PROTECTION, SPRAYERS, VINE GROWING

ВЪВЕДЕНИЕ

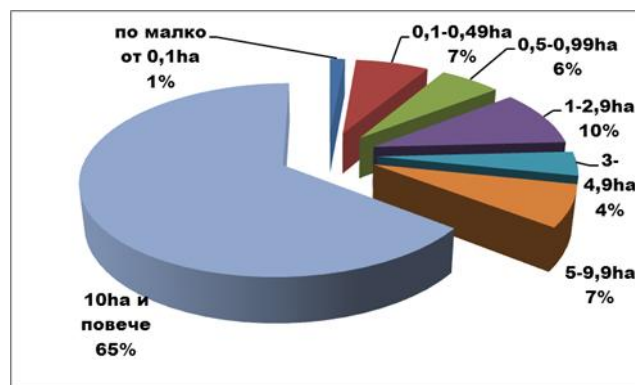
По данни на отдел „Агροстатистика” към МЗХ, през 2016 г. площите с лозови насаждения в земеделските стопанства в България са 50 892 ha. Това е с 0,4% повече в сравнение с предходната година. Около 3% от площите в стопанствата са млади, невстъпили в плододаване лозя, а новозасадените площи с лозя са 790,7 ha, като 520,9 ha са предназначени за производство на бели вина и 269,8 ha – за производство на червени вина и розе (Agrostatistics, 2017).



Фиг. 1. Процентно разпределение на стопанствата, отглеждащи лозя в България, според техния брой

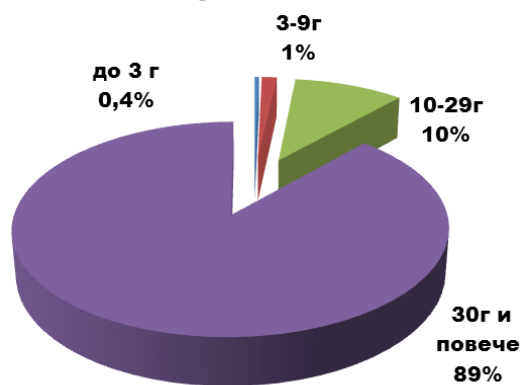
На фиг. 1 и фиг. 2 е показано процентното разпределение на стопанствата, отглеждащи лозя в България, съответно според техния брой и според размера на обработваните площи. Размерите на стопанствата са от един декар до няколко хиляди декара.

Вижда се, че 2/3 от стопанствата, отглеждащи лозя в България имат размер на обработваните площи над 10 хектара. Останалите групи имат от 3 до 10%, а стопанствата, които имат площ от 0,1 хектар и по-малко – са около 1%.



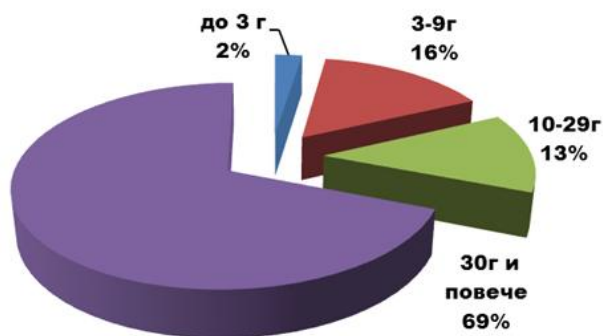
Фиг. 2. Разпределение на стопанствата отглеждащи лозя в България, според размера на обработваните площи

На фиг. 3 и фиг. 4 е показана възрастовата структура на лозята в България, съответно според броя на стопанствата и според размера на обработваните площи. Вижда се, че около 89% стопанства отглеждат стари лозя над 30 години.



Фиг. 3. Възрастова структура на лозята в България, според броя на стопанствата

По отношение на площите 69% от площта е заета с лозя на 30 години и повече, а млади лозови насаждения до 3 години са само 2%.

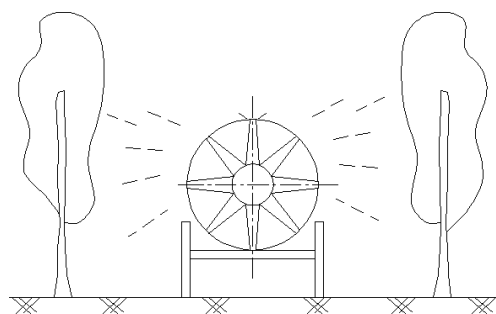


Фиг. 4. Възрастова структура на лозята в България, според размера на обработваните площи

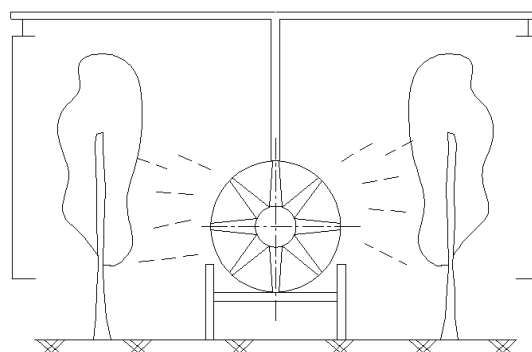
Лозарството в България се отличава с изключително разнообразие от размери на стопанствата, формировки на лозовите растения, междуредови разстояния и организацията на територията в лозовия масив. Във връзка с това, целта на изследването е:

Да се анализират различните методи на разпръскване и транспортиране на работната течност и установят качествените показатели, при използване на различни технологични варианти и машини.

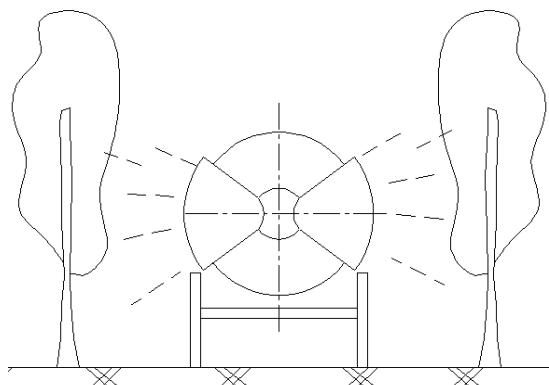
За да се постигне целта, са изследвани половин, едно и многоредови пръскачки, използващи различни принципи на разпръскване. Изследването е проведено при нормално развита листна маса с оптимален разход на работна течност за дадения тип машина, при оптимална производителност на машините.



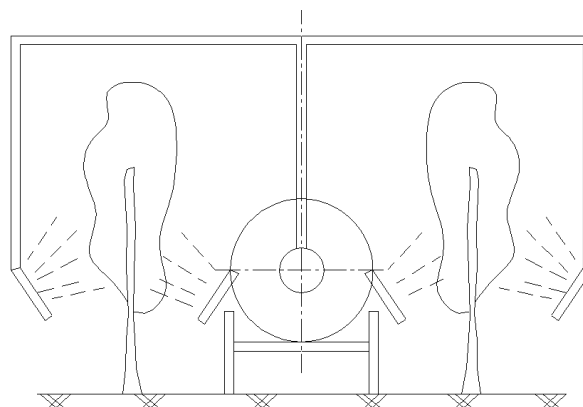
1. Вентилаторна пръскачка Перла-9



2. Вентилаторна пръскачка Перла-9 с екран



3. Пневматична пръскачка Перла-5

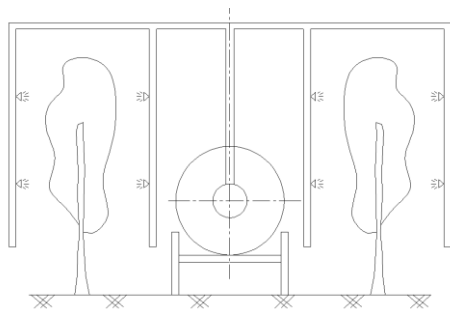


4. Вентилаторна пръскачка Перла-1300

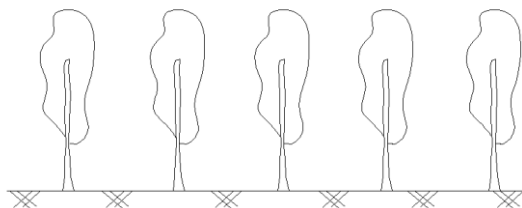
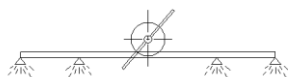
Проучен е въпроса за намаляване на замърсяването на околната среда при вентилаторните пръскачки чрез използването на събиращ екран, улавящ непопадналите капки върху листната маса. Изхождайки от богатото разнообразие от машини и различните размери и формировки на насажденията, е направено сравнително изследване на различните типове машини по отношение на качеството на работа. В това изследването, освен тракторни пръскачки са включени и гръбни, поради наличието у нас на дребни производители от 0,05 до 1,5-2 ha (Prisadashki et al., 1972). Тъй като някои от големите масиви понякога се опръсхват с въздушна техника (авиационни пръскачки) (Boyadjev et al., 1968) също са включени в изследването.

РЕЗУЛТАТИ ОТ ИЗСЛЕДВАНЕТО

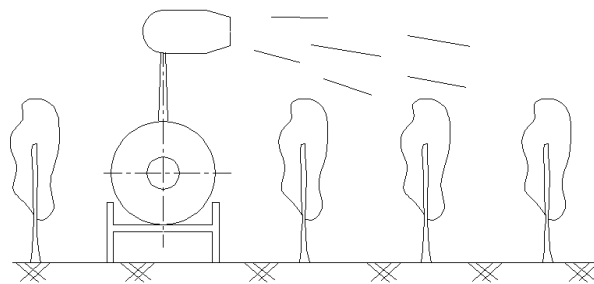
При технологиите, използвани при растителна защита на лозовите насаждения, най-често получаването на капки става чрез изтичане на работната течност през специални накрайници – дюзи (разпръсквачи) (Mortev et al., 2017, Mortev et al., 2018).



5. Щангова пръскачка ПНЩ



7. Самолет Z-37



6. Вентилаторна пръскачка за дрифтово пръскане Теснота-1500

Фиг. 5. Изследвани варианти на технологии и машини за растителна защита

С развитието на технологиите по опазване на лозята се изясниха и някои важни параметри на капковото покритие, на които трябва да отговаря то с цел постигане на оптимална защита на растението. С оглед осигуряването на тези параметри в оптимално къси срокове, един до три дни при възможните почвено климатични условия с максимално опазване на околната среда са разработени многобройни конфигурации пръскачки (Prisadashki et al., 2017).

Съгласно изискванията от британския съвет по растителна защита, за борба с болестите и неприятелите по лозята диаметър на капките трябва да бъде около 150 микрона (Kostadinov et al., 2016). А според изследванията на молдавските учени, най-добре се задържа бордозелов разтвор с капки с размер 150 микрона (Prisadashki, 1980). За маната по лозата е установено, че капковото покритие трябва да се нанася на гръбната страна на листата, там където атакува болестта. За получаване на добро покритие върху листната маса, науката и широката практика показват, че капките трябва да се носят от въздушен поток с параметри от 3-5 до 15-20 m/s. Минималната скорост на въздушния поток е тази, която успява да задвижи листната маса да обърне част от листата и позволи проникването на капките. Максималната допустима скорост се основава на принципа да не се получават разкъсвания на листата, повреди по гроздовете и лоторастите и слепване на листата едно с друго, което също затруднява проникването на капките.

Най-широко приложение в лозарството са намерили четири принципа на разпръскване, при които работят и други машини за растителна защита: хидравличен, пневматичен, комбиниран и механичен (Prisadashki et al., 2017; Bojadjev & Prisadashki, 1971).

В зависимост от предназначението, техническото ниво и използваната работна течност, се прилага един или друг принцип. Най-разпространен е хидравличния принцип, поради простата конструкция на разпръсквачите. Същността на този принцип на разпръскване се състои в подаване на работна течност под налягане от 1-6 MPa в разпръскващ крайник –

специално профилиран при дефлекторните разпръсквачи, завихряща камера при центробежните разпръсквачи, и крайник осигуряващ ударна струя в струя или преграда. Главен недостатък се явява необходимостта от комплектуване на машините с помпа за по-високо налягане и подходящи тръбопроводи, високи изисквания към работната течност, невъзможност да работи с много малки разходи на работна течност и невъзможност за осигуряване на монодисперсен факел, бързо износване на разпръсквачите и помпата което води до нестабилност на установената норма на работна течност и ниска експлоатационна надеждност на машината.

Пневматичния принцип е по-малко разпространен в лозарството за защита на растенията. Тук разпръскването се получава с помощта на кинетичната енергия, която се предава на работната течност от въздушен поток (Bojadjev & Prisadashki 1968). Съществуват два варианта на работа с въздушен поток: с голяма скорост и малък дебит и въздушен поток с малка скорост и голям дебит. Принципа на работа с малък дебит е слабо разпространен в лозарството, където се работи с голям обем. Главното предимство на този принцип се явява липсата на необходимост от помпа с високо налягане, по-ниски изисквания към работната течност и възможност да работи с ниски разходи на течности и фино разпръскване на течността, което обуславя повишена технологична и техническа надеждност.

Най разпространен в лозарството е комбинирания принцип на разпръскване, при който се използват осови вентилатори с голям дебит на въздух и хидравлични центробежни разпръсквачи, работещи с помпи за високо налягане 1,5-4,0 MPa. По този начин се постига дребнокапково пръскане и чрез въздушния поток, който разклаща листата на лозовото насаждение се осигурява равномерно разпределение на препарата от двете страни на листата.

Четвъртия принцип – механичният е разпространен основно през последните години. Неговото внедряване стана възможно благодарение на все по-масовото използване на хидро-, пневмо- и електрозадвижването в селскостопанското

машиностроене (Kostadinov et al., 2011). Разпръскването в този случай се осигурява с подаване на работна течност на въртящ се диск чаша или барабан при обороти от 1500 min^{-1} до 20000 min^{-1} , в зависимост от конфигурацията и диаметъра на разпръскващия орган (Dimitrova et al., 2017).

Главното предимство на този принцип се заключава във възможността за осигуряване на монодисперсен опръскващ факел с контролируем диаметър на капките, в зависимост от режима на работа. Той позволява да се работи при ниски разходи на работна течност, а при комплектуване в един конструктивен възел с осов вентилатор осигурява сравнително добро разпределение на работната течност. Известна трудност на настоящия етап от внедряването му се заключава в отсъствието на подходящи препарати, които се прилагат с минимални дози, а също и в голямото замърсяване на околната среда, възникващо при наличие на въздушни течения. Машините, работещи с този принцип се използват за микрообемно (ултрамалообемно) пръскане. Работна течност в този случай е маслен разтвор или минерално маслена емулсия. При използване на водни суспензии се добавя вещество подобряващо прилепването на пулверизираната течност към листната маса. Капките в този случай не се изпаряват и много добре прилепват.

В зависимост от приетия принцип на разпръскване, режима на работа и метода на транспортиране на капките при дадена конструкция съществуват три вида опръскващи факли: микрообемен (ултрамалообемнен), малообемен и обикновен (Prisadashki, 1980). В предела на режима на някои факли разпръскването може да се обезпечава по два или три принципа. Така например, вискодисперсия факел на разпръскваната течност, освен за микрообемно пръскане, може да осигурява и малообемно при работа с максимална разходна норма. Същото се отнася и за дребнокапковия опръскващ факел, който при ниски норми осигурява малообемно, а при високи норми – обикновено пръскане.

В зависимост от метода на транспортиране на капките до обработвания обект, машините за растителна защита се делят на два типа: с инерционно и с пневматично. В повечето

случаи такова разпръскване се осъществява от машини с щангови опръскващи системи – при тях разпръсквачите се разполагат на щанги. Тъй като кинетичната енергия на капките е недостатъчна за преодоляване на разстоянието от разпръсквачите до опръсквания обект без значителни загуби от изпарения или въздушни течения, разпръсквачите трябва да се монтират на щанга в максимална близост до опръсквания обект. Преимуществено тези машини работят основно с хидравлични разпръсквачи, и по-рядко с пневматични или механични.

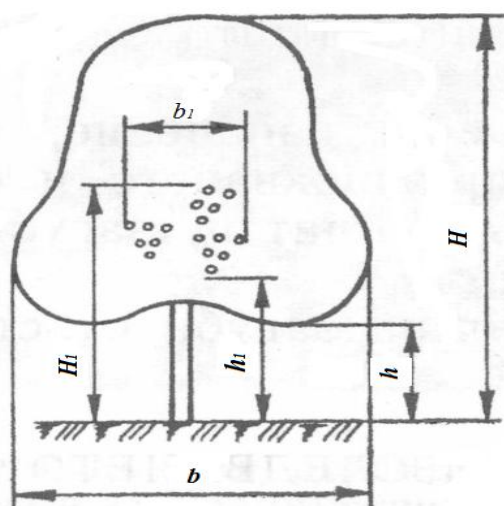
Главен недостатък на щанговите пръскачки се явява голямата маса и размера на опръскващите щанги, което затруднява маневреността на агрегата и в известна степен скоростта на придвижване. Тези опръскващи системи се използват главно за внасяне на хербициди и по-рядки случаи за борба с болести и неприятелите. Те изискват голям разход на работна течност: от 500 l/ha до 1500 l/ha за италианските машини; от 300 l/ha до 1000 l/ha за френските и от 400 l/ha до 1500 l/ha за българските.

Независимо от високия разход на работната течност, качеството на опръскване на тези машини е незадоволително, особено в сравнение с работата на пневматичните.

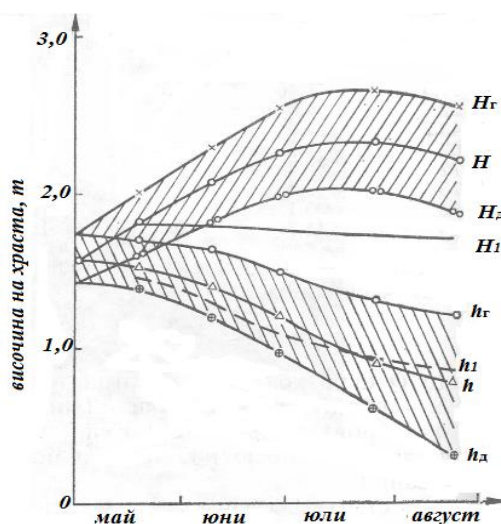
Старите лозя, които са по формировки Омбрела и Мозер, са характеризирани със следните параметри на храста, в зависимост от развитието му (фиг. 5).

На фигура 6 са представени стойностите на плътността на листната маса през периода на растеж и плододане на лозята при различни формировки на лозовите храсти.

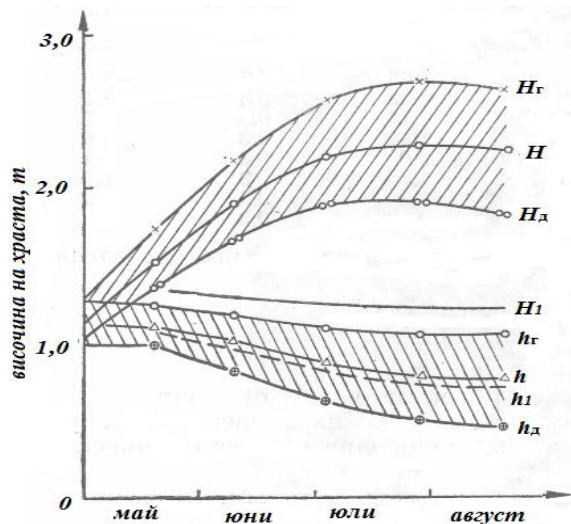
Средната плътност на листната маса е $6 \text{ m}^2/\text{m}^2$ за Омбрела и $4,2$ до $4,7 \text{ m}^2/\text{m}^2$ при Мозер. В зависимост от развитието плътността на листната маса измерена в m^2/m^2 при последните пръскания се увеличава пет-шест пъти спрямо първите, а общия и обем около 10 пъти. Параметрите на лозовия храст нарастват от 4 до 6 пъти. Това изисква растително защитните машини да осигуряват опръскващ факел със съответната плътност.



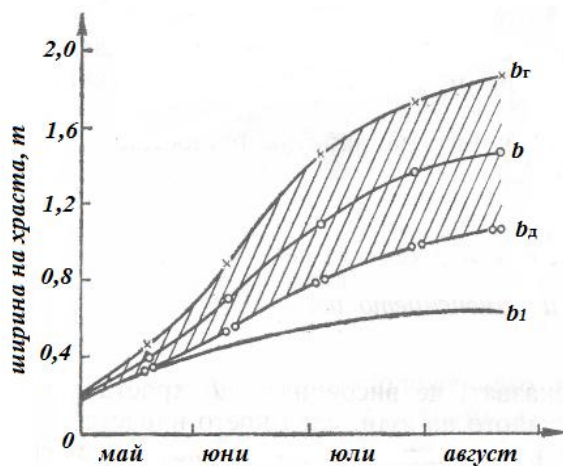
5.1. Параметри на лозовия храст: H - височина; H_1 - височина от почвата до ягоридата (гроздовете след опрашване преди узряване); b - ширина; b_1 - ширина на разположението на ягоридата.



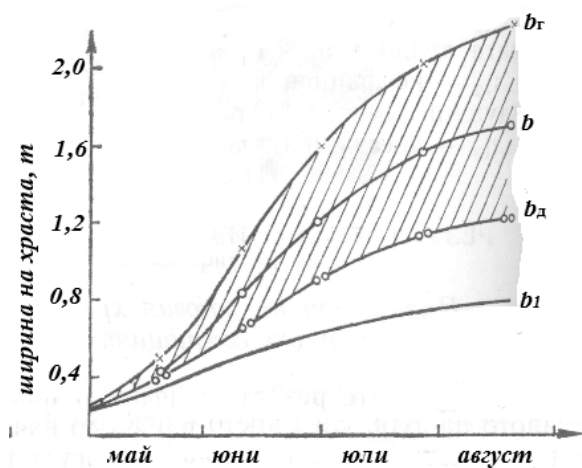
5.2. Изменение параметрите на лозовия храст по височина при формировка "Мозер"; $H_г$, H , $h_г$, $h_д$ – горна и долна граница на параметрите.



5.3. Изменение параметрите на лозовия храст по височина при формировка "Омбрела"

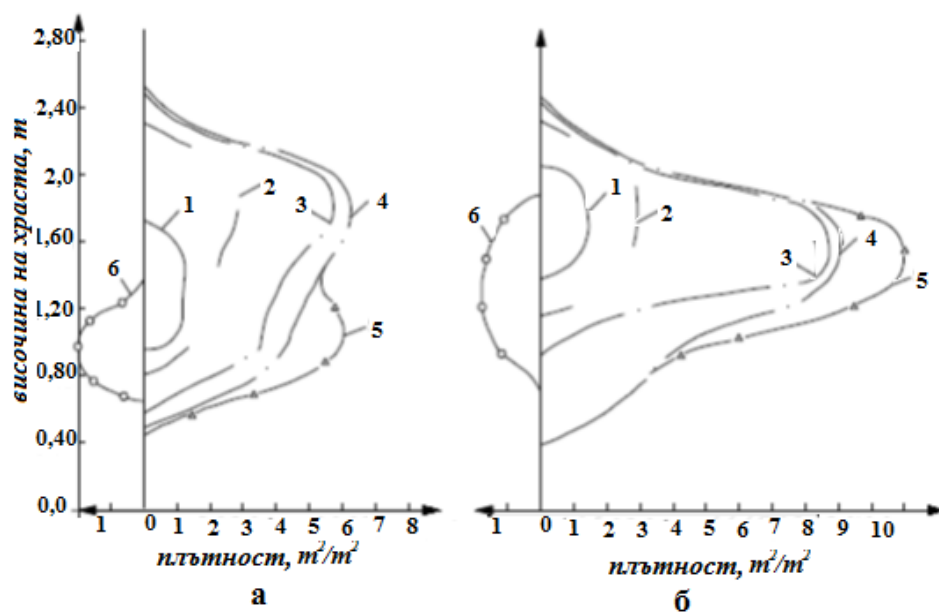


5.4. Изменение параметрите на лозовия храст по ширина при формировка "Мозер"; $b_{Г}$ и $b_{Д}$ – горна и долна граница на параметрите.



5.4. Изменение параметрите на лозовия храст по ширина при формировка "Омбрела"

Фиг.5. Изменение на параметрите на лозовия храст и разположението на гроздето в зависимост от растежа им.



Фиг. 6. Разпределение на плътността на листната маса и ягоридата по височина на храста: а - при формировка Мозер, б – при формировка Омбрела, 1-20 май; 2-13 юни; 3-5 юли; 4-22 август; 5-22 август - листана маса + ягорида; 6-22 август - ягорида

КАЧЕСТВО НА ОПРЪСКВАНЕ

Качеството на опръскване се оценява с „процент на нормално опръсканата площ“ Резултатите от изследването са дадени в табл.1. По процент на нормално опръсканата площ за

горния пояс всички пръскачки дават от челната страна опръскване $90 \pm 5\%$, само самолета дава 75% и дрифтово пръскане с пръскачка Технома-1500 – 70%. По отношение на гръбната страна, най-незадоволителни са ПНЦ и самолета 35% в останалите три е от 56 до 95%.

Табл. 1. Процент нормално опръскана площ, % и оперативна производителност, ha/h

Разположение на пробното листче		Марки машини								
пояси	страна	1	2	3	4	5	6	7	8	9
горен	горна	93	95	93	90	90	90	70	75	85
	долна	95	60	90	20	56	75	25	35	25
среден	горна	78	95	98	90	92	75	100	80	87
	долна	75	53	85	22	73	75	55	40	30
вътрешен	горна	60	95	92	87	80	90	70	60	75
	долна	67	85	82	40	80	90	22	30	25
долен	горна	98	40	93	90	93	90	80	85	55
	долна	83	73	90	64	92	55	70	50	23
Оперативна производителност хектари за час		5,58	5	2,5	4,5	5	0,14	3,2	20-60	0,4

1. Вентилаторна пръскачка Перла-9; 2. Вентилаторна пръскачка Перла-9 с екран; 3. Пневматична пръскачка Перла-5 разход 200l/ha; 4. Щангова пръскачка ПНЦ разход 1000 l/ha; 5. Вентилаторна пръскачка Перла-1300 разход 9 l/ha; 6. Гръбно моторна пръскачка Solo Port разход 150 l/ha; 7. Вентилаторна пръскачка за дрифтово пръскане Теснота-1500 разход 300 l/ha; 8. Самолет Z-37 разход 200 l/ha; 9. Гръбна лозарска пръскачка.

За средния пояс опръскването на горната част на листата е от 75 до 100% а в долната част на листата то 53 до 85% като щанговата пръскачка е 25 до 30% и самолета 40%.

Във вътрешния пояс и зоната на гроздето най-незадоволително е пръскането от долната страна на листата при щанговата пръскачка 25-40%, Технома 22%, и самолета 30%. При тези пръскачки явно трудно ще се опази гроздовата реколта както при оидиума така и при маната. Използването на екран при Перла-9 значително подобрява опръскването в този пояс с 25-35%.

В долния пояс използването на екран влошава качеството, но при добре поддържани лозя този ефект може да се намали. Същото може да се получи и с допълнително регулиране на опръскащата система съобразно оформянето на лозовия храст.

Спектъра на пулверизация

Спектъра на пулверизация, т.е. 80% от течността се опръсква при щанговите пръскачки едрокапково в капки над 150µm. При всички други е дребнокапков от 50-250 µm и само при Перла-1300 е високо дисперсен от 25 до 125 µm. При Перла-5 (пневматична пръскачка) при изследвания разход 200 l/ha спектъра на пулверизация е между високодисперсен и дребнокапков което означава от 25 до 250 µm.

Производителност

Производителността е дадена за час оперативно време – времето за пръскане и времето за завои на машините. Тази производителност най-добре характеризира показателите на машините. Производителността за час производително време е по-малка и зависи много от конкретните условия: време за зареждане на машините за отиване до мястото за зареждане, времето за технологическо и техническо обслужване и други в зависимост от техническите подробности. От таблица 1 се вижда че пръскачката Перла-9 има достатъчна производителност и качество на опръскване, но зъмърсява значително околната среда и има нужда от съвременно оформяне на вентилаторния възел, с което ще се намали разстоянието от между разпръскачите и опръсквания обект.

Полезно ще бъде използването на събирателен екран. Необходимо е да се изследва по-подробно зависимостта на събираната течност от развитието на листната маса и лозовия храст. Предполага се, че това би довело до значителна икономия на пестициди, без да се увеличава много стойността на машината.

Използването на самолет не дава достатъчно качество и трудно би предпазило лозята от силно маносни години. Използването на самолети крие голяма опасност за околната среда и нанасяне на значителни загуби върху опрашващите насекоми (Kostadinov et al., 2016). Авиационното пръскане е забранено от закона за защита на растенията и трябва да се прилага при изключителни случаи, предвидени в закона.

Стопани на лозя до 0,5 ha които нямат трактори могат да ползват гръбно-моторни пръскачки при максимален разход на работна течност от 100-150 l на хектар. Ползването на обикновени гръбни пръскачки само при малки площи до 0,05-0,1 ha комплектувани с един разпръскач с диаметър 1 mm при разход 300-500 l/ha.

ИЗВОДИ

1. Щанговите пръскачки и самолетното пръскане могат да намерят ограничено приложение, само в случаите когато не се изисква покритие на листната маса откъм гръбната страна. Авиационното пръскане може да се използва само в разрешения от закона случай.

2. Тракторните вентилаторни пръскачки трябва да работят с разходи от 200-300 l/ha, а пневматичните – с разход 150-200 l/ha, те може да се комплектуват с улавящ екран който намалява разхода на работна течност до 30%. Трябва да продължат изследванията с използването на улавящ екран като допълнително приспособление към вентилаторните пръскачки.

3. Щанговото и дрифтовото пръскане е незадоволително.

4. Вентилаторни пръскачки „Перла-1300“ могат да работят с разход 90-150 l/ha при ултрамалообем режим и с разход 100-200 l/ha при малообем режим.

5. За лозя до 0,5 ha могат да се ползват гръбно-моторни пръскачки при разход 100-150 l/ha и обикновени гръбни пръскачки с разход 300-500 l/ha.

6. При малки площи до 0,1 ha могат да се използват обикновени гръбни пръскачки при диаметър на дюзите 1 mm и разход 300-500 l/ha.

7. Използването на улавящ екран позволява да се намали процентът на загубената течност чрез улавяне на част от нея и намаляване на замърсяването на околната среда.

8. Улавящия екран при вентилаторните пръскачки вследствие на отразяването на пръскващия факел повишава качеството на работа и увеличава работната ширина от един ред на два реда.

9. Улавящия екран при вентилаторните пръскачки при работа без вентилатор не подобрява качеството на работа на най-важните вътрешен и долен пояс и не трябва да се използва въпреки, че в горния и среден пояс подобрява качеството на работа.

10. При вентилаторните машини с малък дебит на въздушния поток с увеличаване на налягането на работната течност над 0,5 MPa, води до намаляване на уловената работна течност.

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AUTOMATED SYSTEM FOR CONTROLLING THE PRODUCTIVITY OF A SMALL FEED PREPARATION UNIT

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Резюме: Показана актуальность повышения эффективности работы малогабаритного кормоприготовительного агрегата путем автоматизации регулирования его производительности. Приведены результаты разработки соответствующей микропроцессорной системы, с регулированием потока кормового материала при помощи типовой заслонки с индивидуальным электроприводом. Регулирование производится по току двигателя привода дробилки. Представлена функциональная схема предлагаемой автоматической системы регулирования. Определены показатели системы регулирования необходимые для разработки алгоритма управления системой. Приведены результаты лабораторных испытаний работоспособности системы автоматики. Проведены сравнительные производственные испытания дробилки при ручном и автоматическом регулировании загрузки. Показано, что система автоматического регулирования существенно влияет на равномерность загрузки дробилки, улучшает энергетические показатели агрегата, повышает производительность и качество дробления.

КЛЮЧЕВЫЕ СЛОВА: МАЛОГАБАРИТНЫЙ КОРМОПРИГОТОВИТЕЛЬНЫЙ АГРЕГАТ, МИКРОПРОЦЕССОРНАЯ СИСТЕМА РЕГУЛИРОВАНИЯ ПРОИЗВОДИТЕЛЬНОСТИ, ФУНКЦИОНАЛЬНАЯ СХЕМА, ПОКАЗАТЕЛИ СИСТЕМЫ РЕГУЛИРОВАНИЯ, ЭКСПЕРИМЕНТАЛЬНЫЕ ИССЛЕДОВАНИЯ.

Abstract: The urgency of increasing the efficiency of a small size feed-processing plant by automating the control of its productivity is shown in this article. The development results of the corresponding microprocessor system are presented, with the control of the feed material flow by means of a typical damper with an individual electric drive. The control is based on the current of drive motor crusher. The functional scheme of the proposed automatic control system is presented. The indicators of the control system necessary for the development of the control system algorithm are determined. The results of laboratory tests on the operation of the automation system are given. Comparative production tests of the crusher with manual and automatic load control were carried out. It is shown that the automatic control system significantly influences the uniformity of the crusher loading, improves the power parameters of the plant, productivity and quality of crushing.

KEY WORDS: SMALL SIZE FEED-PROCESSING PLANT, MICROPROCESSOR CONTROL SYSTEM OF THE PRODUCTIVITY, FUNCTIONAL SCHEME, INDICATORS OF THE CONTROL SYSTEM, EXPERIMENTAL STUDIES.

1. Введение

Обеспечение населения продукцией животноводства является главной задачей агропромышленного комплекса Республики Казахстан. В связи с этим особое внимание обращается на проблемы животноводства и кормопроизводства. Важнейшим условием успешного развития животноводства является создание прочной кормовой базы, совершенствование средств автоматизации процесса производства и переработки кормов. В технологии автоматизации сложных объектов и процессов, имеющих важное народнохозяйственное значение сложилась устойчивая тенденция к использованию интеллектуальной системы управления. Применение автоматической системы регулирования производительности позволит повысить эффективность использования кормоприготовительного агрегата и снизить эксплуатационные затраты на приготовления кормовых смесей у производителей сельскохозяйственной продукции.

В Республике Казахстан используются кормоприготовительные машины различных фирм. Единственным отечественным разработчиком и изготовителем такого рода машин является ТОО «Казахский научно-исследовательский институт механизации и электрификации сельского хозяйства» (КазНИИМЭСХ).

Малогабаритный кормоприготовительный агрегат ДУ-11 (рисунок 1), на основе которого проводились исследования, предназначен для измельчения всех видов стебельных и зерновых кормов, кукурузных початков. К недостатку машины при измельчении зерна относится необходимость, ручного регулирования его производительности при помощи заслонки, который сужает или расширяет щель, через который поток зерновых материалов поступает на рабочий орган. Ручное регулирование, как правило, приводит к неполной загрузке или перегрузке агрегата. При этом, как показывает практика эксплуатации этих агрегатов, большую часть рабочего

времени электропривод работает в недогруженном состоянии, и наоборот нередко случаи отключения электропривода из-за перегрузки двигателя. Приведенное кроме необходимости постоянного наблюдения за работой машины, влечет за собой снижение энергоэффективности и машины качества продукции.

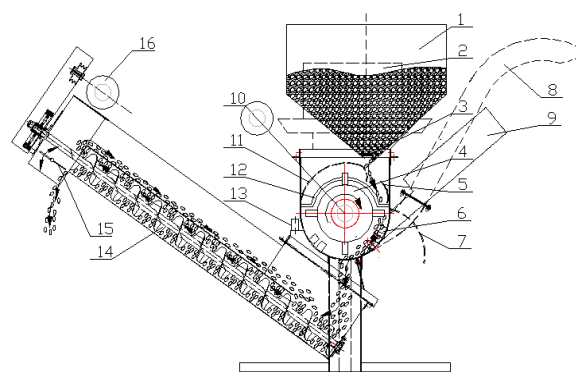


Рисунок 1 – Схема универсального малогабаритного кормоприготовительного агрегата

1 – бункер зерна; 2 – бункер сена; 3 – заслонка;
4 – ротор молотковый; 5 – вихрь;
6 – дека; 7 – отражатель; 8 – дефлектор; 9 – лоток; 10 –
электродвигатель ротора;
11 – отсекаль; 12 – лифтер; 13 – рамка; 14 – шнек-
смеситель; 15 – заслонка шнека; 16 – электродвигатель
шнека-смесителя.

Целью работы является повышение эффективности использования универсального малогабаритного кормоприготовительного агрегата, путем разработки системы автоматики позволяющей обеспечить регулирование производительности агрегата за счет

изменения потока кормового материала поступающего на рабочий орган машины.

Цель достигается путем встраивания в агрегат индивидуального электропривода заслонки регулирующей загрузку молоткового ротора. При этом регулирование загрузки будет производиться по току двигателя привода молотковой дробилки.

Результаты работы позволят повысить технический уровень и конкурентоспособность рассматриваемого малогабаритного кормоприготовительного агрегата и будут использованы в конструкции других аналогичных кормоприготовительных машин и агрегатов.

2. Экспериментальные исследования.

Работы по разработке систем автоматического управления для повышения производительности измельчителей и дробилок используемых в различных отраслях промышленности проводятся учеными и инженерами ряда стран.

Например, известны мини мукомольная машина (1) и молотковая мельница с устройством управления подачей (2). Выполнены работы по повышению эффективности процесса дробления путем разработаны ПИД-регуляторы дающих возможность быстрого отслеживания параметров процесса (3), регулирования работы дробилки в соответствии с заданным законом (4), разбивки процесса дробления на ряд событий при математическом моделировании (5,6,7), осуществления контроля характеристик исходного материала (8), учета время зависимых переменных в контроле дробилок и динамического управления конусными дробилками (9,10), динамической компенсации износа футеровки дробилок (11).

Также, существует интеллектуальная система управления грубой дробилкой угля на основе разработанного алгоритма сравнения изображения предварительно обработанного материала с заданным изображением, разработана автоматическая система с электронным датчиком для измерения давления в порошковой камере дробилки, непосредственно передающий цифровую информацию на управляющий компьютер, также известна интеллектуальная система управления конусной дробилкой для улучшения стабильности и динамических характеристик системы, способ он-лайн оптимизации конусных дробилок позволяющей максимизировать общую производительность дробилок путем регулирования скорости при неизвестных заранее изменениях характеристик (12,13,14,15).

В серийно выпускаемых мини-дробилках Auto Loader, Hopper, Sandvik предусматриваются системы автоматической сигнализации нехватки материала, системы автоматического настраивания дробилки к условиям подачи, защита двигателя.

В соответствии с заданной целью и с учетом вышеуказанных работ, нами была разработана система автоматического регулирования (16). Функциональная схема системы регулирования приведена на рисунке 2. В рабочей части механической характеристики асинхронного двигателя ЭД1 рабочего органа РО, ток пропорционален величине крутящего момента на его валу и следовательно нагрузке рабочего органа агрегата. Значение тока через датчик ДТ, осуществляющего обратную связь в системе, передается микропроцессору МП, который, сравнивая его с заданными значениями тока в соответствии с алгоритмом работы системы регулирования, передает соответствующий управляющий сигнал пусковому устройству ПА2 электродвигателя ЭД2 задвижки бункера ЗБ. В результате последняя открывается или закрывается регулируя поток корма из бункера к рабочему органу и соответственно изменяя ток двигателя ЭД1. Пусковая аппаратура ПА1 служит для управления двигателем ЭД1, индикационный прибор ИП показывает текущее значение тока двигателя и режимы работы агрегата.

При разработке системы автоматики необходимо учитывать время пуска двигателя дробилки. Нами на натуральном образце универсального малогабаритного кормоприготовительного агрегата были получены пусковые диаграммы тока и скорости двигателя при пустом барабане дробилки и наличии остатка 2 и 4 кг зерна (ячмень) в барабане (рисунок 3). Записи проводились при помощи осциллографа DSO 3064, шунта и тахогенератора ТЧ-10Р.

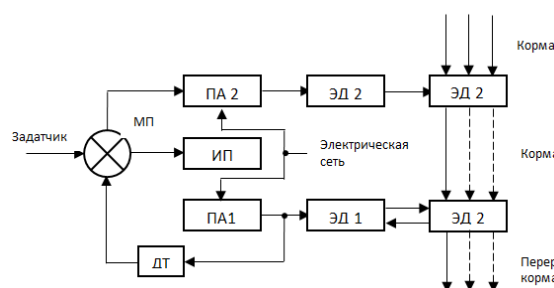


Рисунок 2 – Функциональная схема системы регулирования МК – микроконтроллер; ИП – индикационный прибор; ДТ – датчик тока; ПА1, ПА2 – пусковая аппаратура; ЭД1 – электродвигатель рабочего органа; ЭД2 – электродвигатель задвижки; РО – рабочий орган; ЗБ – задвижка бункера.

При пустом барабане дробилки электродвигатель загружен только моментом сопротивления холостого хода рабочего органа. Из осциллограмм видно, что в этом режиме, время пуска приводного двигателя дробилки составляет $t_n = 2,0$ с.

При наличии в барабане дробилки остатков зерна также происходит надежный запуск. При этом, при остатке в бункере 2 кг ячменя $t_n = 3,1$ с, при остатке в бункере 4 кг ячменя $t_n = 6,0$ с.

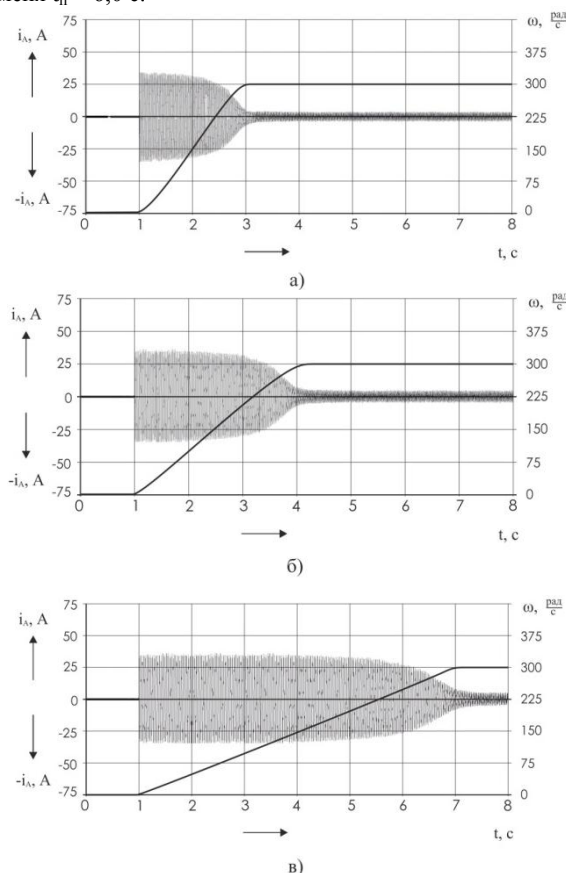


Рисунок 3 – Осциллограммы мгновенных значений тока и скорости двигателя при пуске а) при пустом барабане дробилки; б) при наличии остатка 2 кг ячменя в барабане; в) при наличии остатка 4 кг ячменя в барабане.

При разработке системы регулирования, важное значение имеет выбор величины диапазона регулирования тока, определяемого значениями минимально I_{min} и максимально I_{max} допустимых токов. При узком диапазоне, возможна реакция системы управления на незначительные кратковременные изменения нагрузки, при широком – длительная работа в недогруженном режиме. В то же время, величина наименьшего допустимого тока I_{min} ограничивается снижением производительности машины и энергетических показателей электродвигателя, наибольшего I_{max} – нагревом обмоток электродвигателя.

Энергетическая эффективность двигателя, в основном определяется зависимостью коэффициента полезного действия (КПД) η двигателя от его коэффициента загрузки κ_z , $\eta = P_2/P_{2н}$, где P_2 и $P_{2н}$ соответственно текущая и номинальная мощности на валу двигателя. При коэффициенте загрузки $\kappa_z = 0,875$ КПД асинхронного двигателя дробилки достигает наибольшего значения $\eta = 0,88$. Целесообразно, чтобы значение I_{min} соответствовало этому значению коэффициента загрузки. Ток I_{max} по условиям нагрева, не может быть больше номинального тока двигателя I_n соответствующего $\kappa_z = 1$, следовательно $I_{max} = I_n$.

Таким образом, $I_{min} = 20,8$ А и $I_{max} = 23,65$ А. Среднеарифметическую величину найденных токов принимаем равным значению рабочего тока $I_p = 22,1$ А.

Чтобы исключить реакцию системы управления на быстротечные выбросы нагрузки, в программе системы регулирования предусмотрено вычисление среднего значения амплитуды тока за одну секунду и использования этой величины при последующих шагах программы.

Для проверки возможности реализации предлагаемого способа регулирования, были разработаны алгоритм и программа управления, макетный образец системы, стенд для его испытания и проведены испытания (17). Принципиальная электрическая схема стенда, показана на рисунке 4.

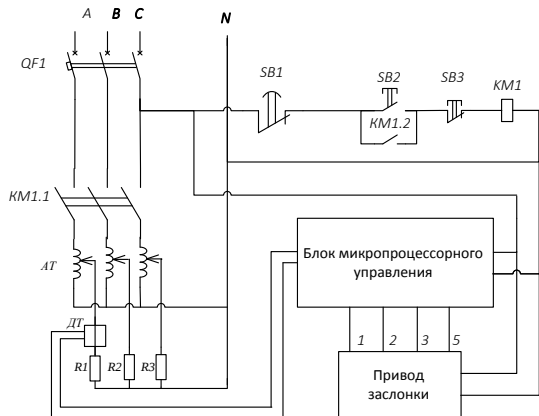


Рисунок 4 – Принципиальная электрическая схема лабораторного стенда

В системе автоматического регулирования были использованы контроллер МК типа Arduino Mega построенный на микроконтроллере ATmega2560 сопряженный с блоком согласования с дисплеем LCD-LCD12864, реле типа LC1 D4011, датчик тока трансформаторного типа до 100А SCT013, привод заслонки BELIMO SM 230 ASR с встроенным датчиком угла поворота заслонки.

Изменение тока двигателя дробилки имитируется с помощью трехфазного автотрансформатора АТ, к которому подключена активная нагрузка R1-R3. Фазный ток нагрузки и угол поворота заслонки записывались при помощи осциллографа DSO 3064.

Для лабораторных испытаний в программу были заложены следующие действующие значения токов – $I_{min} = 4$ А, $I_{max} = 10$ А, $I_p = 7$ А.

Результаты опытов приведены на рисунке 5.

В начальном положении задвижка закрыта. При увеличении тока, задвижка постепенно открывается и при достижении током рабочего значения $I_p = 7$ А останавливается, открывшись на 63° (рисунок 5а). Далее, при изменении значения тока в интервале от $I_{min} = 4$ А до $I_{max} = 10$ А задвижка остается неподвижной и сохраняет свое положение (рисунок 5б). При значении тока более $I_{max} = 10$ А, задвижка начинает закрываться и при уменьшении тока до $I_p = 7$ А останавливается, закрывшись до 44° (рисунок 5в). При значении тока менее $I_{min} = 4$ А, задвижка начинает открываться и при увеличении тока до $I_p = 7$ А останавливается открывшись на 70° (рисунок 5г).

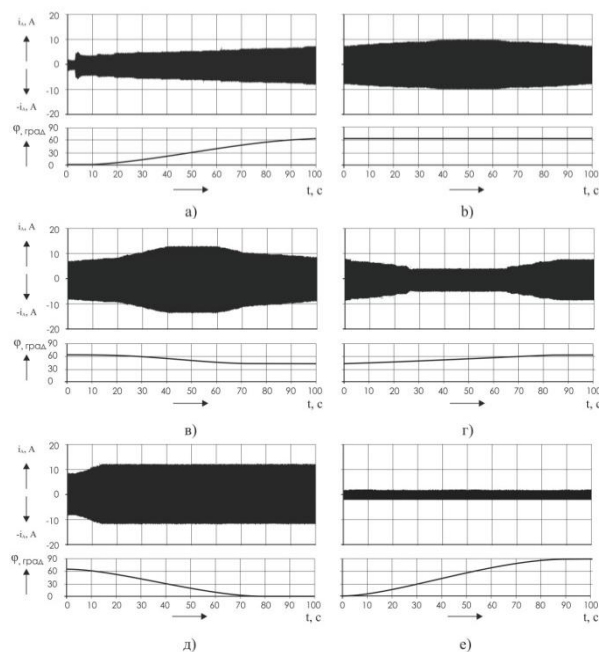


Рисунок 5 – Осциллограммы мгновенных значений тока двигателя и положения заслонки

При постоянном значении тока превышающим $I_{max} = 10$ А, задвижка, не зависимо от начального положения полностью закрывается (рисунок 5д) и при постоянном значении тока ниже $I_{min} = 4$ А, задвижка полностью открывается (рисунок 5е).

Данный эксперимент подтвердил работоспособность макетного образца системы регулирования в соответствии с заданным алгоритмом.

Далее нами были проведены производственные испытания экспериментального образца агрегата с автоматической системой регулирования производительности.

Составлена принципиальная электрическая схема соединений электропривода машины с учетом возможности ручного и автоматического управления процессом дробления (рисунок 6).

зерен. Ручное регулирование не обеспечивает должную равномерность загрузки. При уменьшении загрузки дробление зерна чрезмерно увеличивается, и наоборот, при перегрузке слишком снижается.

4. Заключение

Сравнение результатов экспериментов, полученных при ручном и автоматическом регулировании загрузки дробилки, показывает, что система автоматического регулирования существенно влияет на равномерность загрузки дробилки, улучшает энергетические показатели агрегата, повышает производительность при дроблении зерна на 0,4 т/час и качество дробления.

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THE JUSTIFICATION FOR COLTER SPREADER CONFIGURATION SELECTED FOR BROADCAST SEEDING AND MINERAL FERTILIZER APPLICATION

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Abstract: The aim of this work the graphical regularities of the fertilizer pellets outflow through a flattened cone have been considered in an effort to justify the configuration of universal colter spreader which would provide even distribution of the grass and grain seeds, and mineral fertilizer pellets within the soil. It has been determined that to achieve the granular materials' uniform outflow in the direction transverse to travel speed over the colter spreader surface its elliptical cross-section is required. To this effect the section plane should be directed perpendicular to travel speed, cross over all cone generators at an angle between it and central axis of the latter other than $\pi/2$.

Keywords: colter, broadcast seeding, mineral fertilizers, spreader, grass sowing, fertilizer applicator.

Research analysis shows that broadcast subsoil seeding method creates the most favorable conditions for growth and development of cultivated crops [1, 2, 3]. This is due to more even distribution of seeds over the field than that in the row seeding method [4, 5]. Thus, at broadcast seeding method the yield capacity of corn crops and grass at an average increases by 10...30% if compared with close-drilling and row seeding methods [6, 7, 8].

At broadcast seeding the best quality is assured by the seeding machines equipped with tine colters having spreaders for broadcast subsoil seeding which evenly distribute the seeds throughout all field area [9,10]. Passive spreader of various configurations is the most common and simple in design. However, such spreaders are characterized by highly irregular distribution of seeds and fertilizer pellets by the grasp width [11,12]. Vibrating spreaders create low irregularity of the seed distribution but their design is more complicated.

It is known from analysis of various methods of positional application of fertilizers in the soil that in order to minimize the fertilizer absorption by soil and to create conditions for the plants' better access to them in risky agriculture environment of North Kazakhstan the using screen method for local application of the main fertilizer dose for grain crops to 10...15cm depth of permanent moisture.

To implement the said technology various distribution devices and attachments are applied. However, the existing distribution devices for internal soil application of mineral fertilizers equipped with passive spreaders do not provide the required (up to 15%) regularity of solid fertilizers screening [13].

In connection therewith, the work aimed to improve the regular distribution of various crop seeds and fertilizer pellets over the field area by means of improving the universal spreader design in under-tine space is of great topicality and practical significance.

Goal of research is to justify the configuration of universal cone – colter spreader providing even distribution of grass seeds, grain crops and mineral fertilizer pellets within the soil.

Research objective:

- research of regularities of fertilizer pellet outflow through the flattened cone.

Materials and methods of research.

According to operational hypothesis we have suggested, in order to achieve the even band distribution of fertilizers the impact is required to stabilize the law of mineral fertilizer distribution at outlet from the colter shank and in under-tine space of the colter [14].

If using the technical solution [15], which consists in that at the cone surface cutting off in directions transverse to movement, in parallel to the cone generator, the bottomscatterer edge has the parabolic curve shape and oval shape in the plan (in this case the material traces 1^1-2^1 , 2^1-3^1 , 3^1-4^1 may be equal, figure 1), then it is possible to provide even distribution of the fertilizer pellets across the width of colter.

To realize the suggested hypothesis, for subsoil application of fertilizers the new colter design has been developed. The structure consists of fertilizer funnel shank to which the duck-foot tine or tine point is connected (at grass seeding). In the base of fertilizer funnel shank the spreader cone is fixed.

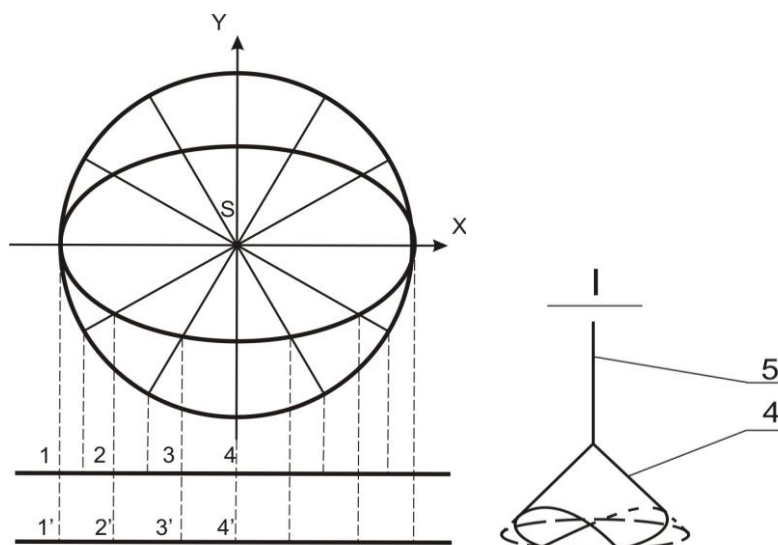


Figure 1 – Cone scatterer, top view

Essential prerequisite to achieve the even fertilizer pellets flowing down from cone is their falling into top part of the flattened cone. The spreader cone purpose consists in even distribution of materials in transverse direction of under-tine space.

Research results and their analysis. Figure 2 represents the scatterer cone with S vertex and $S-1, S-2, \dots, S-12$ generators. Three projections, namely: front, horizontal and profile are shown in the figure. If the travel speed direction $\vec{U}$ is taken by Ox axis then in profile projection the $S''' - 7''' - 1'''$ scatterer cone moves towards us, the lookers. Flowing down the cone generator the fertilizer particles fall into the $1'', 2'', \dots, 6'', 7''$ points. In the figure the fertilizer traces are shown by dash-dotted lines. As it is seen from the figure, the interval between the application traces are not equal and have the sizes of δ_1, δ_2 u δ_3 , though the fertilizers were delivered to the scatterer cone by equal angular spacing δ° .

The task is to make equal the unequal intervals δ_i between the traces of fertilizer particles drilling. For this purpose, we divide the cone scatterer base diameter D into six parts with equal

intervals Δ . The project traces of fertilizer particles drilling are reconstructed through the obtained 1, 2, ...7 points. For differentiation they are drawn by dash lines. Let us determine the points of obtained drilling traces intersection with corresponding generators in the profile projection:

$$\begin{aligned} Cl.1 \cap S'''1'' = L_1''; & \quad Cl.5 \cap S'''5'' = L_5''; \\ Cl.2 \cap S'''2'' = L_2''; & \quad Cl.6 \cap S'''6'' = L_6''; \\ Cl.3 \cap S'''3'' = L_3''; & \quad Cl.7 \cap S'''7'' = L_7''; \\ Cl.4 \cap S'''4'' = L_4''; & \end{aligned} \quad (1)$$

Through the obtained points connecting by smooth lines we obtain the profile projection of unknown curve $L_1'', L_2'', \dots, L_7''$, which can provide regular outflow of the fertilizer particles over the scatterer cone surface in the transverse direction z . However, its form and the way to obtain it remain unknown.

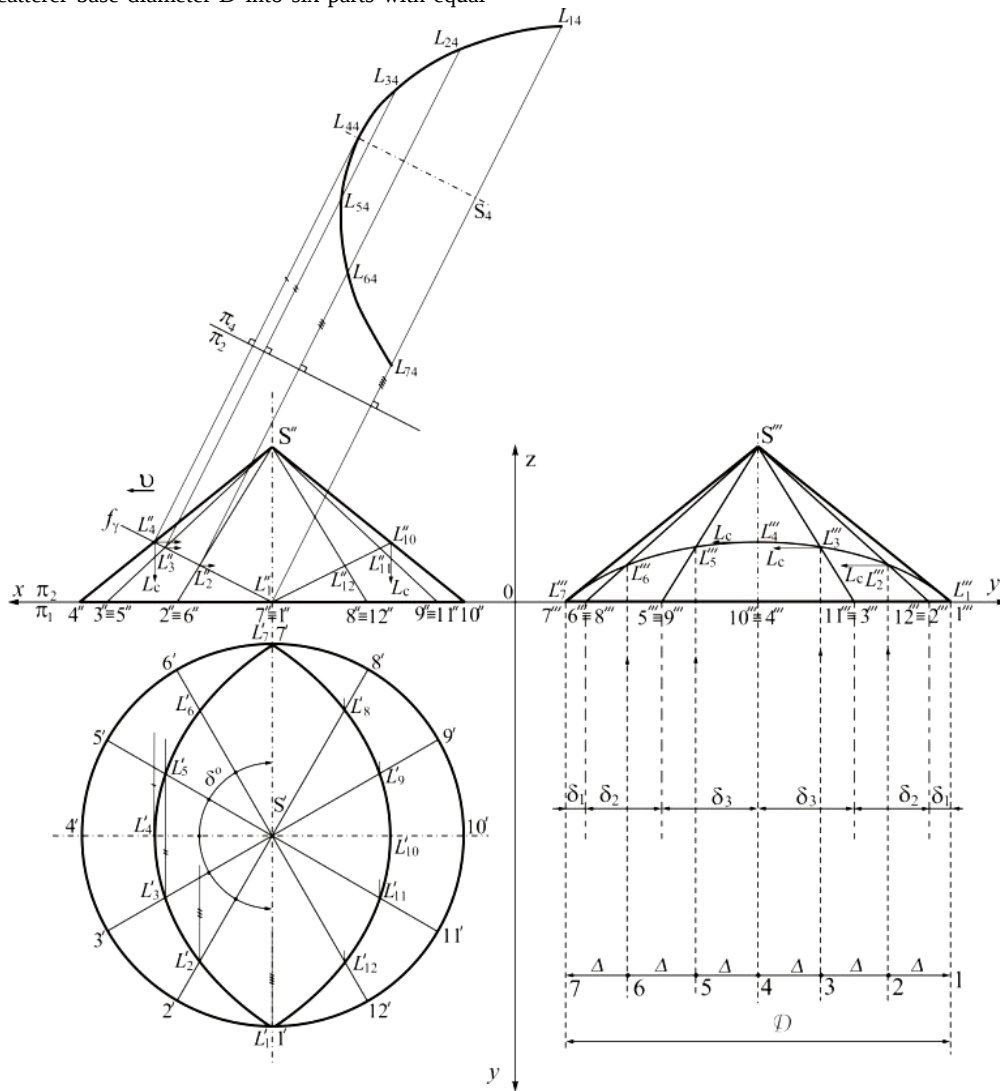


Figure 2 – To rationale for the spreader cone configuration

To determine the form of, and way of obtaining the desired curve let us project the $L_1'', L_2'', \dots$ points, as well as L_4'' point located at the curve peak to corresponding generators of front projection. As it is seen from the figure, L_1'' remains in

$1''$ point. In the cone backside L_7'' coincides with it. The remained points are determined at connection line (L_s) intersections with corresponding generators:

$$\begin{aligned} \mathcal{I}cL_2''' \cap S''2'' &= L_2''; \\ \mathcal{I}cL_3''' \cap S''3'' &= L_3''. \end{aligned} \quad (2)$$

Having $L_1'' - L_2'' - L_3''$ connected we obtain a straight line. It goes without saying that this line constitutes a front trace of frontally projecting plane γ , on which the desired curve is located. Further we determine:

$$f_\gamma \cap S''4'' = L_4''.$$

Back projection gives a point at the peak of desired curve:

$$\mathcal{I}cL_4'' \cap S'''4''' = L_4''.$$

Cone is the surface symmetric with respect to its central axis. Therefore, the L''_{10} , L''_{11} and L''_{12} are determined according to symmetry law.

Horizontal projection of the desired line points is determined by means of the found front projections to corresponding cone generators on horizontal plane of the projection:

$$\begin{aligned} \mathcal{I}cL_2'' \cap S'2' &= L_2'; \\ \mathcal{I}cL_3'' \cap S'3' &= L_3'. \end{aligned} \quad (3)$$

.....

However, the obtained projection does not give the comprehensive idea of the curve nature since it consists of two branches, at that, each branch is not represented in full.

To determine the natural form of desired curve let us use the descriptive geometry methods for projection transformation,

particularly, the method of projection plane replacement. For this purpose, let us replace π_1 plane of initial system by π_4 plane of the new system. At that, the replacing plane should be parallel to γ plane, as the desired curve lies on it:

$$(L_1, -L_2, -\dots) \frac{\pi_2}{\pi_1} \frac{\pi_2}{\pi_4} (L_{14}, -L_{24} - \dots) \parallel \pi_4. \quad (4)$$

Based on the provision that coordinates in the replacing plane should be equal to coordinates in replaced plane, we determine the points L_{14} , L_{24} , L_{34} and L_{44} . The desired figure is symmetric with respect to $S'4'$ generator. That is why the points L_{54} , L_{64} and L_{74} are determined according to the symmetry law.

Upon the obtained points connecting by smooth lines one may note that the desired curve constitutes a half of ellipse the major axis of which is equal to $b = L_{14} - L_{74}$, and minor axis is equal to $a = 2L_{44} - S_4$.

To confirm the reached conclusion let us consider the α cone with S vertex and rotation axis i . Some γ - plane intersects all its generators but is not perpendicular to the cone axis. Thus:

$$\gamma \cap \alpha = l,$$

some curve line is obtained, figure 3.

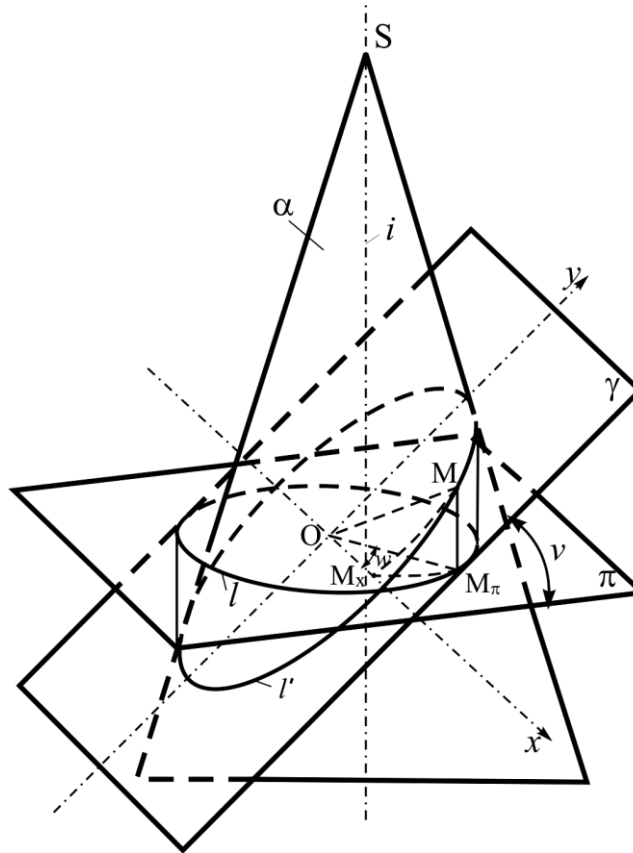


Figure 3 – To rationale for the spreader cone configuration

Let us project all points of l curve to π plane which is perpendicular to i axis of the cone. For example, the M point is projected to M_π point.

The given cone is circular, therefore, on the π plane the circle l' of radius a will be obtained. Its value is equal to the radius of cone on the plane π .

Let us draw through the center of circle l' the coordinate axis OY in the γ plane and axis OX perpendicular to it. M point coordinates on these axes are equal to:

$$\begin{aligned}x &= OM_x; \\y &= MM_x.\end{aligned}\quad (5)$$

We take following notations:

V – acute angle between the planes γ and π ;

w – angle between radius of circle $a = OM_\pi$ and Ox axis;

M_x – point M projection to Ox axis.

Let us determine the M point coordinates via w parameter.

From the triangle $OM_x M_\pi$:

$$\frac{OM_x}{OM_\pi} = \cos w; \quad OM_x = a \cos w. \quad (6)$$

From the triangle $M_x M M_\pi$:

$$\frac{M_x M_\pi}{M_x M} = \cos V; \quad M_x M = \frac{M_x M_\pi}{\cos V}. \quad (7)$$

From the triangle $M_x O M_\pi$:

$$\frac{M_x M_\pi}{OM_\pi} = \sin w; \quad M_x M_\pi = a \sin w. \quad (8)$$

Substitute (8) into (7):

$$M_x M = \frac{a \sin w}{\cos V}. \quad (9)$$

In (9) take notation

$$\frac{a}{\cos V} = b; \quad M_x M = b \sin w. \quad (10)$$

From (6) and (10) with due regard to (5) we obtain:

$$\begin{aligned}x &= a \cos w; \\y &= b \sin w.\end{aligned}\quad (11)$$

So, we have obtained parametric equation of ellipse. Therefore, the line l' constitutes the ellipse, as was to be proved. Minor axis of ellipse is equal to a , and major axis is equal to b ; and it is protruded in direction of axis y .

Conclusions and proposals for production.

To obtain the regular outflow of the mineral fertilizer pellets and other granulated materials over the surface of spreader cone of experimental colter in direction transverse to the travel speed its elliptical cross-section is required. To this effect the section plane should be directed perpendicular to travel speed,

cross over all cone generators at an angle between it and central axis of the latter other than $\pi/2$.

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FIELD TESTS OF AN AUTONOMOUS FIELD ROBOT FOR WIDE-ROW CULTIVATION

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Abstract: In Industrial Institute of Agriculture Engineering (Poznań, Poland), together with the Institute of Vehicle of Warsaw Technical University and firm Promar Poznań action have been taken to design autonomous field robot for sowing and cultivation of crops. Designed robot is to be autonomous device. It will automatically perform the sowing, weeding and selective spraying plant crops such as sugar beet or maize. The robot will work in various terrain and weather conditions. It will move on the dirt roads and on cultivated fields, and therefore it will have to overcome mud, sand, puddles and avoid obstacles such as ruts, bumps in the road and stones. The achievement of this objective required the development of a suitable chassis, system of autonomous control in terms of driving and realization of agronomic processes.

KEYWORDS: FIELD TESTS, AUTONOMOUS ROBOT

1. Introduction

Care for the natural environment and health benefits of crops and the growing interest of consumers in tasty food encourage farmers to look for ecological agricultural production technologies. The threat to the environment and the quality of agricultural produce is primarily the excessive use of herbicides. The need to maintain high yields and low production costs requires alternative methods of plant protection against excessive weed infestation. A simple and effective method of destroying weeds is their mechanical removal. It should be remembered that weeds are competition for arable crops and the longer they occur in the plantation, the greater the losses they cause in the crop. Manual weeding e.g. on large plantations sugar beets, due to the high cost of labor and the problem of employing a sufficient number of employees to perform the work at the required time, effectively discourages growers from ecological farming.

For mechanical removal of weeds hoes with tools passive, active or a combination of them are used. These machines are aggregated with an agricultural tractor and can be equipped with automatic guidance systems for working elements. Usually, they allow you to remove weeds only from inter-row surfaces. More advanced cultivars, called intelligent ones, locate desired plants using GPS signals (using knowledge of the position of sown seeds) or image processing systems. These are solutions in the testing phase, with limited use. Problems occur with obtaining the required accuracy of the location based on the GPS system, especially since the sowing and weeding operations are performed in separate operations often by various tool carriers. Location systems based on image analysis are unreliable with big weed infestation, which may occur, for example, at delayed emergence, as well as due to bad weather conditions that prevent entry of the tractor on the subsoil at the required time. The solution to these and similar problems can be found by application of a common tool carrier in the form of an autonomous field robot. Its construction (power demand, stability, maneuverability, way of hanging the tools) should be adapted to various field treatments. An important requirement is also that it is a light machine, with good traction properties, causing low soil degradation and capable of performing operations in difficult terrain conditions (Fig. 1).

The introduction to the widespread use in agriculture, horticulture and forest nurseries of autonomous field robots may be a breakthrough in the current approach to large-scale crops, where this type of production could not do without the use of herbicides. The possibility of using autonomous field robots will increase the availability of organic food products and should lower their prices.

2. The concept of the designed robot

At the Industrial Institute of Agricultural Engineering in Poznań, in cooperation with the Institute of Vehicles of the Faculty of Automobiles and Working Machines of the Warsaw University of Technology and with PROMAR Sp. z o.o. (private limited companies). An attempt was made to develop an autonomous field robot for sowing and cultivating plants.

Three types of systems implemented in this type of machines can be distinguished: driver assistants, automatic control and autonomous, unmanned machines. The last type assumes the possibility of working without an operator (possible supervision). A device with these features must meet a number of requirements and perform several basic work modes, such as: task initiation (mission), task calibration, path planning, track tracking, observing and waiting, avoiding obstacles, detecting mission errors / interrupting missions, ending missions. Some of these tasks are deterministic (e.g. planning and tracking the path), while the part is completely reactionary (avoiding obstacles). The essence of the implementation of individual modes is the appropriate algorithm, while the key information for him is the appropriate measurement.

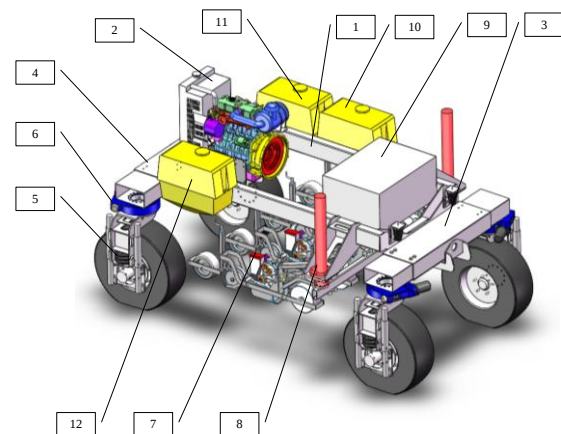


Fig.1. Geometric structure of the initial robot model

- 1 - supporting frame of the traveling platform; 2 - diesel engine;
3, 4 - suspension axle beams; 5 - running wheel assembly;
6 - turntable; 7 - tools (point seeder / active cultivator / sprayer);
8 - system of lifting the tools; 9 - oil tank; 10 - fuel tank,
11, 12 - sprayer tanks

The robot conception used in the project assumes building a device with the highest level of autonomy. The robot will automatically perform sowing, weeding and selective spraying of crop plants such

as sugar beets or corn. The robot will operate in various field and weather conditions. It will move across field roads and arable fields, so it will have to overcome mud, sand, puddles and avoid obstacles such as ruts, road unevenness and stones. Achieving this assumption required the development of an appropriate chassis, control system and algorithms of autonomous control in the field of traction and execution of agro-technical processes.

It was assumed that the robot will have a modular construction [1]:

- carrier

The basic module is a stand-alone carrier cooperating with specialized modules. The carrier will be 4 wheeled vehicle, able to work remotely and autonomously. The drive wheels provide hydraulic motors and planetary gears. This way of transfer of power will get the high ground clearance of the machine (free space for tools) and independent suspension - important when working in difficult terrain. In addition, the speed differentiation of individual wheels and their appropriate setting will result in high maneuverability.

- sprayer

Sprayer will be able to realize during the sowing or weeding dosage of liquid fertilizer or other liquid preparations of plant protection. The application will be executed surfactants (for plants or soil), or to the soil (of the depth below the surface of soil) in a continuous or selective based on the optical identification and location of objects.

- seeder

For precision seeding seeder will be applied with the seed position recording system in the field.

- hoe

The main working unit will be active hoe robot for precise control of weeds in both the inter-rows and row crops.

Particularly noteworthy is a comprehensive approach to the mechanization of wide row crops, including the use of a common autonomous tool carrier for the execution of sowing procedures and the mechanical destruction of weeds, assisted if necessary by the selective use of liquid plant protection products and fertilizers.

Essential part of robot is its control system [2]. The main sensor system is based on a specialized GPS receiver that provides position information with an accuracy of less than 100 mm. This system is mainly used to guide the robot along a designated path, and also (in cooperation with the vision system) for precise sowing - accurate seed location information will be used to build seed maps that will be used as auxiliary information for precise weeding and spraying. The view from the front camera will be used to increase the accuracy of the robot positioning. This will allow you to improve the path accuracy of the robot relative to the rows of plants. The vision system will also be used to detect stationary objects (Fig. 2). At the same time, the second vision unit will also be used to acquire images from the camera directly in front of the active hoe and sprayer. Additional information from acceleration sensors and encoders embedded in the wheels will also be used for navigation purposes. To determine the angular acceleration, the IMU (Inertial Measurement Unit) will be required. This will enable:

- correction of the robot's trajectory,
- precise work of an active hoe,
- position adjustment and precise dosing of fertilizers or plant protection products by the sprayer.

The diagram of the robot drive control system is shown in Fig. 3.

3. Field research of the robot's test model

After the test model of the mechanical structure of the robot was built, tests of its motion properties were carried out. The robot was then equipped with executive elements, including tools for performing agrotechnical works [3], but algorithms of autonomous control were not yet launched. The robot was controlled manually

using a special panel and manipulators (Fig. 4). Status of the robot was visible on a special interface (Fig. 5).

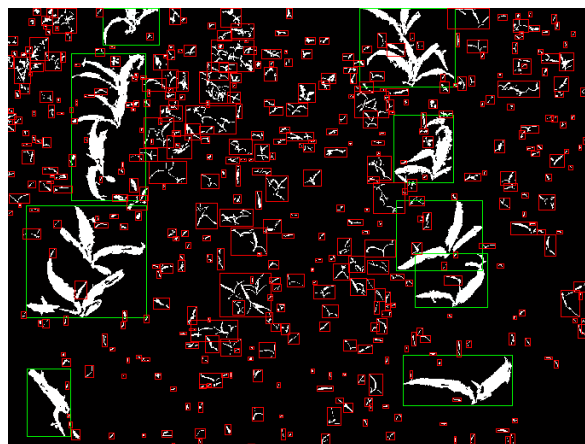


Fig. 2. Tests of the system distinguishing crop plants from weeds

In order to check the correctness of selected solutions and introduce any corrections:

- A) Operating loads occurring under different operating conditions of the robot have been determined.
- B) The hydraulic robot system has been tested. Pressure was measured in selected elements of the hydraulic system, including hydraulic motors driving individual wheels, during maneuvers in various operating conditions of the robot work.

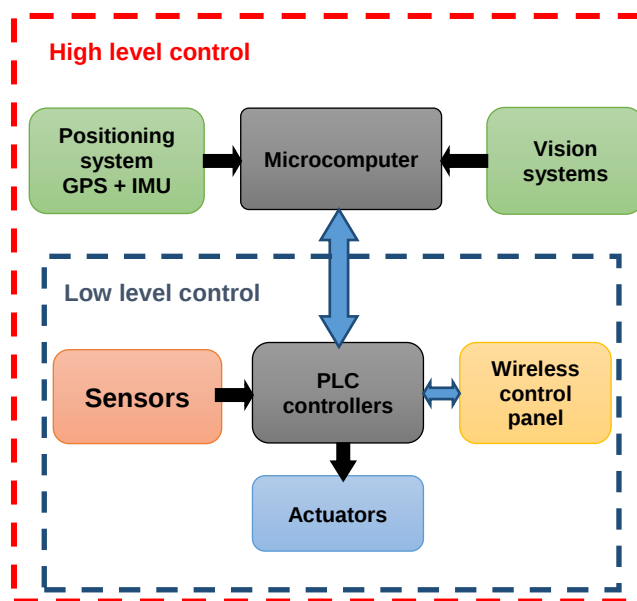


Fig. 3. Diagram of robot driving control system



Fig. 4. The panel used for manual robot control during field tests

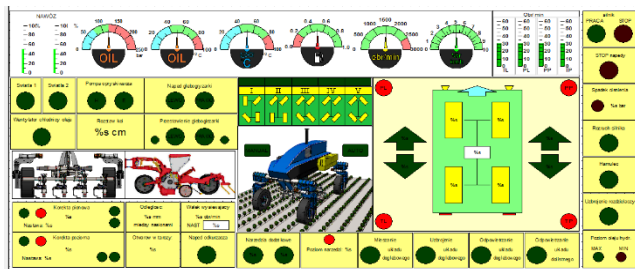


Fig. 5. User interface used during robot tests

- C) Field tests of the robot's driving properties have been carried out (stability and drivability and the possibility of overcoming obstacles). The way of moving the vehicle with various configuration of driving wheels and for various configuration of the cornering method was checked.
- D) Driveability of the field robot's work in terms of communication range, machine reaction and control quality has been checked. The correctness of the wheel control algorithms and the force with which the wheels of the field robot's platform interact on the ground have been checked.
- E) Limit situations were determined for the robot's control system under various operating conditions (various types of ground and different speeds) in the range of obstacles, turns and turns (Fig. 6). During the tests on various types of ground (cultivated field and non-arable field), maneuvers of the running gear were tested for different speed settings. The possibility of moving during difficult conditions was also tested, where the robot's wheels went deep enough to prevent normal operation.
- F) Introductory laboratory and field tests of the work of the tool modules were carried out. The sowing, weeding and spraying processes were tested.

4. Sample results and discussion

This section presents sample results obtained during robot field tests.

The figure 7 shows the behavior of the control system and the actuation system when robot is starting off. The characteristics show

the start-up of the vehicle along with the gradual increase of the speed to the maximum and next rapid shutdown of the drives.



Fig. 6. Field tests of the robot in various operating conditions (various types of ground and different speeds)

At start-up, you can see incorrect supply of current signal to feed pumps (A). This behavior was caused by deliberate tuning of the controllers in the controller for research purposes. The drive pump (hydraulic motors) on right side received working setting about 2 s faster. Due to the initiation of start only one of the parties, the pressure in the engines increased dramatically (B). After starting the second pump, the pressure dropped and began to stabilize at a set level. At increasing the speed to the maximum, pressure remained at a similar level. Interesting is the period when the vehicle moved at maximum speed and encountered harder working conditions, which caused rapid pressure jumps and speed reducing (11:03:40 - 11:03:45). The last fragment of the diagram shows the braking of the machine. After switching off the settings, the pressure on the pumps and motors increases temporarily due to the closing of the valves.

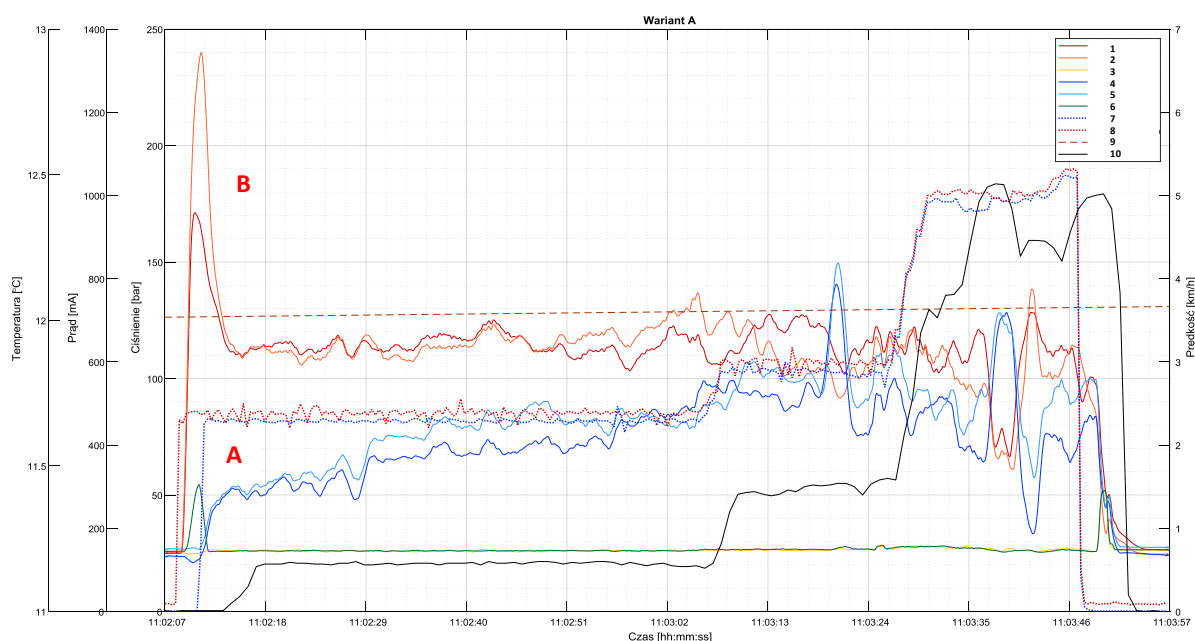


Fig. 7. The working parameters of the robot during acceleration in a straight line

1 - pressure in the right front wheel motor [bar]; 2 - pressure in the right rear wheel motor [bar]; 3 - pressure on the right side motor output; 4 - pressure in the left rear wheel motor [bar]; 5 - pressure in the left front wheel motor [bar]; 6 - pressure on the left side motor output [bar]; 7 - left wheels drive current [mA]; 8 - right wheels drive current [mA]; 9 - temperature [°C]; 10 - robot velocity km h⁻¹

5. Conclusions

The carried out research allowed to remove defects occurring in the robot's structure, e.g. rebuilt the cover of the hoe working in inter-rows, in order to enable proper copying of the area. Also there was modified the hydraulic system, in which there was a harmful difference of pressure on both sides of the robot.

The general course of the tests, however, confirmed the correctness of the adopted structural solutions in the area covered by the tests.

Currently, the next cycle of research on robot work in the autonomous mode is being conducted. The subject of the research and tests are primarily robot control algorithms in the field of intelligent control of working tools and independent moving of the robot through the field.

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INFLUENCE OF DIESEL COMPOSITE FUEL ON THE BASIS OF RICINIC OIL ON INDICATORS OF THE TRACTOR DIESEL OPERATION

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Timeliness: The relevance of the matter is confirmed by numerous researches of the alternate engine fuel, in particular, from the renewable sources of raw materials (biomass) studied in various regions of the Russian Federation and abroad.

For obtaining diesel composite fuel as raw materials are attractive oil crops which are not used in the food industry, for example, carthamus, false flax, colza, winter cress, crambe and castor-oil plant.

At the moment the need for the ricinic oil used in military, chemical, machine-building, radio-electronic, printing, paint and varnish, medical, cosmetic and other industries in the Russian Federation grows. As a result of processing of a castor-oil plant fruits by cold press in residual meal or pod there are about 10% of oil which is possible for allocating by processing of meal on technology of extraction with receiving technical ricinic oil with a possibility of application as raw materials for receiving paint and varnish, lubricants and diesel composite fuel.

However viscosity of ricinic oil is 10 times more, than at diesel gas oil therefore it is necessary to apply in mix with low-viscosity components. As additive to ricinic oil can be used mineral oil fuel and bioethanol due to which mixture viscosity considerably decreases, cloud point and crystallizations temperature decreases. [8]

As diesel composite fuel is investigated mixture of ricinic oil (RicO), and summer diesel gas oil according to National State Standard 305-2013. On the basis of the carried-out calculation are defined indexes of a working cycle, dependences on concentration of components of blenderized fuel are shown.

On the basis of the calculation, are determined the working cycle indexes, are shown the dependences on the concentration of the components of the composite fuel. Based on the results of the calculation, the obtained indicators were compared with diesel gas oil.

KEY WORDS: RICINIC OIL, CASTOR OIL, DIESEL COMPOSITE FUEL, DIESEL, INTERNAL COMBUSTION ENGINE, MIXTURES, POWER, ECONOMY, ALTERNATIVE FUEL, BIOMASS.

1. INTRODUCTION

One of the main ways of development of agricultural and timber production is the partial transfer to alternative fuels from biomass. This is confirmed by the Decree of the President of the Russian Federation of July 7, 2011 No. 899 on priority areas for the development of science, technology and technics in the Russian Federation. The development of alternative energy and the use of biomass are also taken into account by many international studies and scenarios for the development of energy and industry in general.

According to the innovative scenario for the development of world energy until 2050 as a result of the change in the structure of the global fleet of vehicles, as well as the introduction of strict restrictions on emissions of toxic substances, renewable energy and alternative fuels will develop significantly. [2]

An innovative scenario involves overcoming the crisis of the industrial phase (in its postindustrial or non-industrial form) and the crisis of relations with the environment on the basis of the transition to a new phase of development of the society, economy and energy. [2]

As a consequence of this, the energy sources obtained from biomass will receive a great development already in modern times, they can be applied to the vehicles and tractors without significant upgrades of the existing ICE.

For this reason, is being carried out large-scale international scientific research to diversify energy supply sources that will solve the problem of replacing petroleum fuels, significantly expand the resource base for obtaining motor fuels, facilitate the solution of fuel supply issues for vehicles and fixed installations remote from large settlements, can be achieved reduction of production costs. [3,4]

Biodiesel, methane and diesel composite fuel are preferred as alternative energy sources for automotive vehicles in this period.

Biodiesel as a fuel type has a disadvantage expressed by a low temperature of turbidity and crystallization as a result of which it cannot be applied in northern regions where the ambient temperature reaches over -30°C . This type of alternative fuel is more applicable in countries with a warm climate and having smaller reserves of petroleum raw materials than the Russian Federation, such as Brazil, India, Vietnam. [2,6]

Gas-diesel is installed on KamAZ-65115, Scania G340 and NefAZ-5299 buses. But as a result of the insufficient level of gasification of the territory of the Russian Federation, and especially in the rural areas, it is impossible to use methane as an alternative source of energy for tractors in agricultural enterprises and logging enterprises that are remote from the gas pipeline.

Among the preferred types of biofuel applicable to agriculture and forestry is the multicomponent diesel composite fuel produced from the mixing of petroleum diesel fuel and plant oil. [13]

The most studied source of plant raw materials is rape. However, the use of oils from other oilseeds deserves attention. Today, there is a growing need in the Russian Federation for ricin oil in military, chemical, machine-building, radio electronic, printing, paint, medical, cosmetic and other industries. In connection with the current policy on import phase-out and oil demand in the Russian Federation, since the main share of ricinic oil on the market is produced in India. It can be considered that technical ricinic oil is a promising competitor to other plant oils, investigated as components of diesel composite fuel. [14,15,16]

Castor oil plant belongs to the genus of perennial plants of the family of euphorbia. This genus is represented by one species - castor beetle (*Ricinus communis*), which is subdivided into several subspecies.

In tropical and subtropical countries, castor oil is grown as a perennial or annual culture, in areas with a temperate climate - only as an annual one.

In crop rotation, castor oil plant is placed after winter wheat, winter barley, maize and leguminous plants. The castor oil plant itself is a good precursor for spring cereals.

The cultivar of castor oil plant Khortitskaya 1 is mid-season, plants of medium height, weakly-violet, laid a dense brush at 8-10 knots. Bole height is 50-70 cm. Khortitskaya 1 is highly resistant to fusarium, adapted for mechanized harvesting. Oil content of seeds is 51-52%. The yield is 1.2-1.5 t / ha. [17]

As a result of processing of the castor oil plant fruit by cold pressing, only 10% of the oil remains in the residual meal, which can be recovered by processing the meal using the technology of extraction or processing of fruits by hot pressing to obtain a technical ricinic oil with the possibility of using it as raw material for the production of paintwork materials and diesel composite fuel.

2. ESTIMATE INDICATORS OF TRACTOR DIESEL ENGINE WORK

To use diesel composite fuel on the basis of ricinic oil, it is necessary to analyze the influence of the constituent components on the economic, power-cycle characteristics of the diesel engine.

Calculation of the D-240 engine was carried out by the methods of Prokopenko R.M., Khorosh A.I., Bashirova R.M. [9,10,11] using three fuel mixtures: RicO + DO, respectively

Table No. 1 - Elementary composition of mixtures.

№	Indexes	DO	Mixture 1	Mixture 2	Mixture 3
1.	Carbon C	0,870	0,8495151	0,8426868	0,8358585
2.	Hydrogen H	0,126	0,1239453	0,1232604	0,1225755
3.	Oxygen O	0,004	0,0265396	0,0340528	0,041566

From the obtained data it is possible to single out the primary feature of diesel composite fuel based on ricinic oil, the oxygen content

(mixture 1, mixture 2, mixture 3). The results of the studies are presented in Table No. 2.

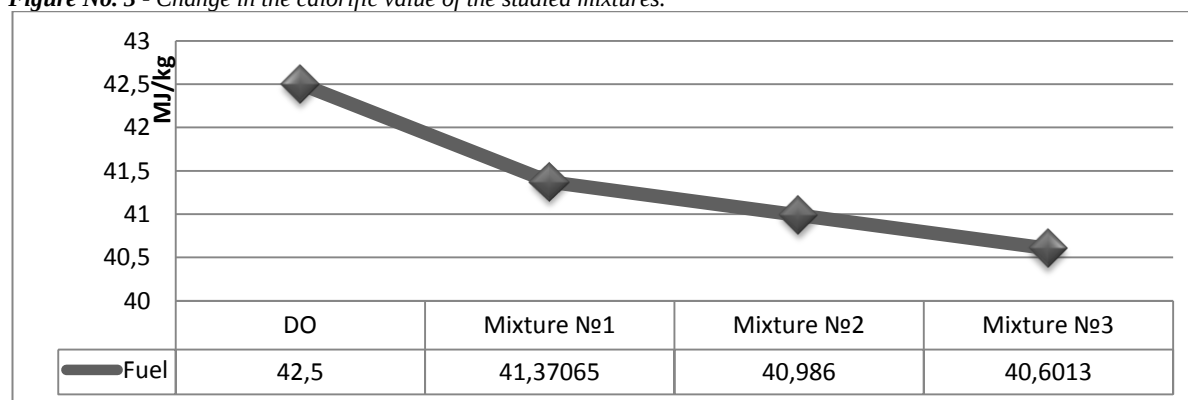
Is determined the content of the basic substances included in the composition of ricinic oil by the method of Egazaryants C.B, and is calculated the elementary composition of the mixture, which is determined by the content of C, H, O by the formula Mendeleyev D.I. The results of the studies are presented in Table No. 1 [12].

increases which favorably affects the completeness of combustion of the mixture.

Table No. 2 - Indicators of the working cycle of a diesel engine using mixtures of different concentrations.

№	Indexes	DO	Mixture 1	Mixture 2	Mixture 3
1	2	3	4	5	6
4.	Heat of fuel combustion, MJ / kg	42,5	41,37065	40,986	40,6013
5.	Theoretical amount of air, kg \ kg of fuel	14,35	13,94	13,80	13,67
6.	Heat of combustion of a inflammable mixture MJ / kg	1.8874	1.8882	1.88876	1.8880
7.	Coefficient of molecular change	1,041	1,04283771	1,04334192	1,043856
8.	Combustion temperature, °K	2158,0	2168,5	2168	2167,5
9.	Mean effective pressure, MPa	0.7233	0.709635	0.71	0,71035
10.	Effective efficiency	0.363	0,351645	0,3517	0,35174
11.	Effective specific fuel consumption, g \ kW	233,3	247,46	249,74	252,08
12.	Effective power, kW	63,0	61,80	61,83	61,86
13.	Power change,%	-	-1,91	-1,86	-1,81
14.	Change in specific fuel consumption,%	-	+6,07	+7,048	+ 8,05

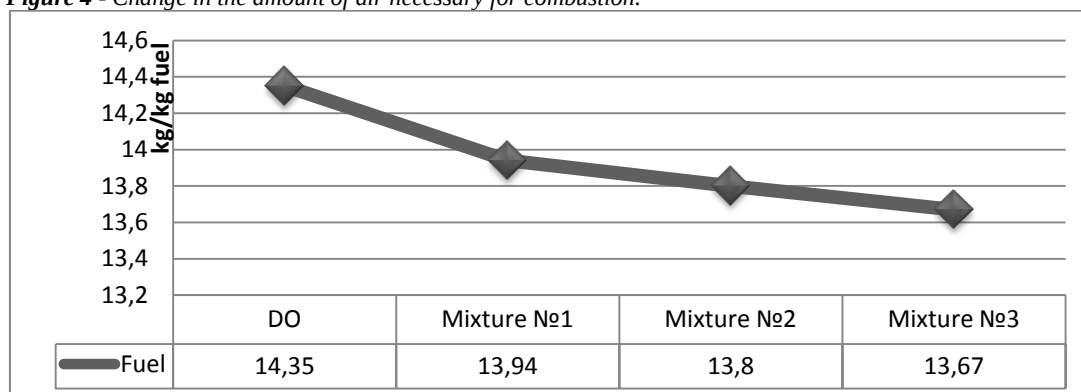
Figure No. 3 - Change in the calorific value of the studied mixtures.



When analyzing the calculations, it is noticed the decrease in the heat of combustion the test mixtures by 2.6% in comparison

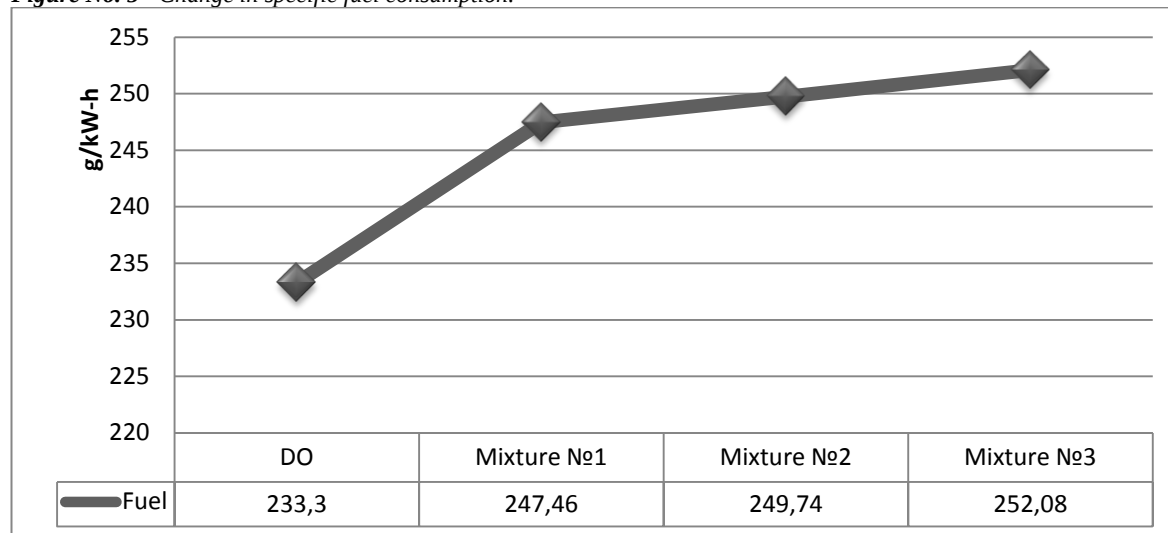
with diesel fuel. With a change in the oil fraction by 5%, the heat value of combustion changes by an average of 0.93%.

Figure 4 - Change in the amount of air necessary for combustion.



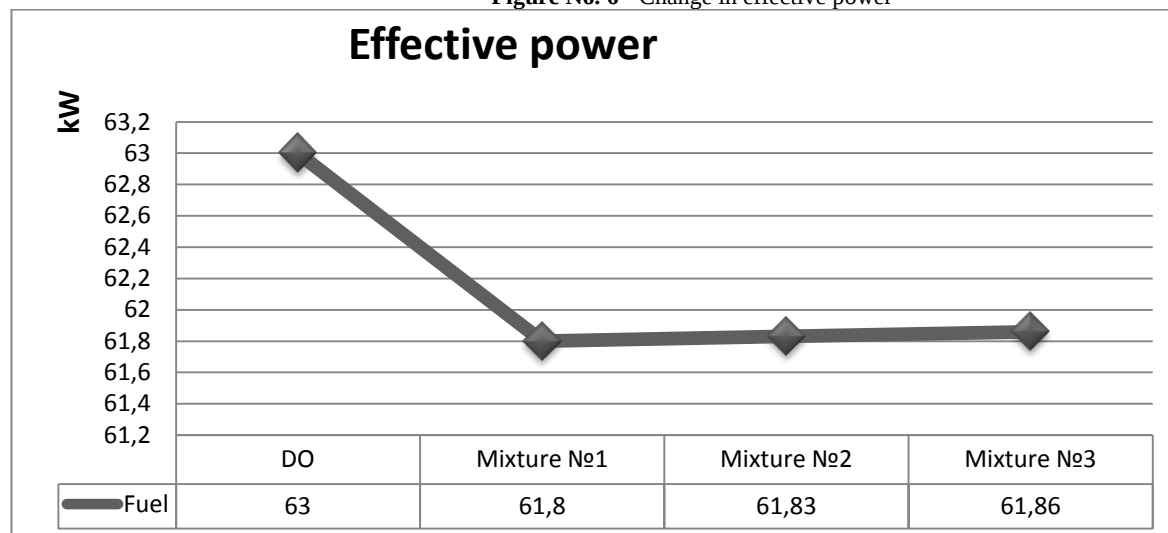
When using mixtures based on ricinic oil, is reduced the required amount of air for combustion of the mixture: according to calculations in comparison with diesel fuel, it decreases by 2.86%, with the required amount of air decreasing as the concentration of oil in the mixture increases. This is due to the high oxygen content according to the elemental composition of $C_3H_5O_3$ ($C_{18}H_{33}O_3$) ricinic oil compared to diesel fuel.

Figure No. 5 - Change in specific fuel consumption.



Specific fuel consumption is significantly increased by 6.07% in comparison with diesel fuel, and is revealed regularity: with an increase in the oil concentration for every 5%, the consumption increases by 2.3 g / kWh.

Figure No. 6 - Change in effective power



There is a slight decrease in power compared to the operation of the engine on diesel fuel mixture: an average of 1.86-1.91%, depending on the concentrations of the used components.

3. CAN BE DONE THE FOLLOWING CONCLUSIONS ON THE BASIS OF THE CONDUCTED STUDIES:

1. The required amount of air for combustion of the mixture decreases as increases the concentration of oil in the mixture.
2. There is slight decrease in the power of engine D-240 at 1.86-1.91% when working on diesel composite fuel based on ricinic oil compared to diesel fuel.
3. The economy of the engine deteriorates by 6.07% in comparison with diesel fuel or by an average of 2.3 g / kWh, with an increase in the proportion of oil in the mixture by 5%.

It was found that the heat of combustion of the combustible mixture is relatively small changed, which is associated with a decrease in the number of moles of combustion products. [5.8]

4. CONCLUSION.

Thus, diesel composite fuel based on ricinic oil has the ability to be used on diesel internal combustion engines, according to the calculations, it has similar indicators with mixtures based on rapeseed, mustard, soybean oil.

These studies will be of interest to companies engaged in the production and processing of castor oil in order to reduce fuel costs. But it makes sense to reduce the share of the oil base and increase the environmental friendliness of the ICE to conduct research on the addition of bioethanol.

5. CONFLICT OF INTERESTS.

The author confirms that the presented data do not contain a conflict of interest.

6. GRATITUDES.

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**THE INFLUENCE OF MODIFICATION MOLDBOARD PLOW ON THE
IMPLEMENT HORIZONTAL FORCE AND PLOWPAN LAYER**

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1. Introduction

Plow pan is a compacted layer of soil that is difficult for roots or water to penetrate. It often forms just below the depth of plowing, when, over the years, repeated plowing orients all of the soil particles in the same a direction and depth, causing a compacted layer of soil. This layer has a value of penetration resistance more than 2 MPa which considered to be the critical value for root penetrate deeply into the soil. Moreover one of the most factor that causing an increase in the soil strength in that layer is the wheel pass running in the plowing furrow during tillage operation.

2. Problem discussion

In the conventional plowing system, the top layer of soil is usually loosened by using a moldboard plow. At a depth just below the plowing layer, about 20 cm below the surface of the soil, a compacted layer develops, it is commonly called the plow pan or hardpan, and it forms at a depth of between of 20 and 35 cm below the soil surface. This dense layer has a thickness of around 10-15 cm. However, this varies from field to field and within fields. Hardpan could be the source of negative influences on the physical properties of soil, such as higher bulk density, lower soil porosity percentage, and higher soil strength when compared to the topsoil and underlying horizontal. This explains its adverse effect on soil properties, plant activity, and land erosion by runoff. The most important external factors that have a direct effect leading to the formation of this layer are the plowing implement action causing the smearing of the soil and also, the tractor wheels running in the opened furrow during plowing operations.

3. Objective and research methodologies

The aim of this study is to modify and develop the moldboard plow design by adding an extra part (chisel tine), that works below the plowing layer immediately after wheel pass on the bottom of the furrow, which has been previously opened in the last run of the plowing operation. This leads to soil loosening , therefore reduced soil penetration resistance.

A mounted type of reversible moldboard plow, in-furrow plowing system was used. The plow was modified by adding an extra component (chisel tine) which was combined to the frame of the plow by the Henryk Batyra Company in Lublin, Poland (Figure1). The tine was working in the center of the wheel pass in the furrow during the plowing operation. The experiment was conducted using the complete randomized block design with three replications. The first treatment was the tine setting (depth and working width below the moldboard plow) with five levels: NO TINE represented the normal plow without an extra tine (control), TINE1D10 and TINE1D15 represented a 5 cm tine width which worked below the plowing layer at a depth of 10 and 15 cm respectively. TINE2D10 and TINE2D15 represented a 10 cm tine width that worked below the plowing layer at a depth of 10 and 15 cm respectively. The second treatment was three forward speeds 1.94, 3.9 and 6.3 km·h⁻¹.

To measure horizontal force of the implement, an adjustable three-point hitch sensoring dynamometer with a horizontal force capacity of 100 kN was used. It was located between the tractor and an implement. It was made by polish company in Poznań, Poland (Figure 3).

To determine soil penetration resistance, Eijkelkamp penetrolgger was used after wheel pass on the furrow (before and after tine action), at three levels of soil depth below the plowing layer 0–5, 5–10 and 10–15 cm, that corresponded to a soil depth of 20–25, 25–30, 30–35 cm from the ground surface (Figure 2)

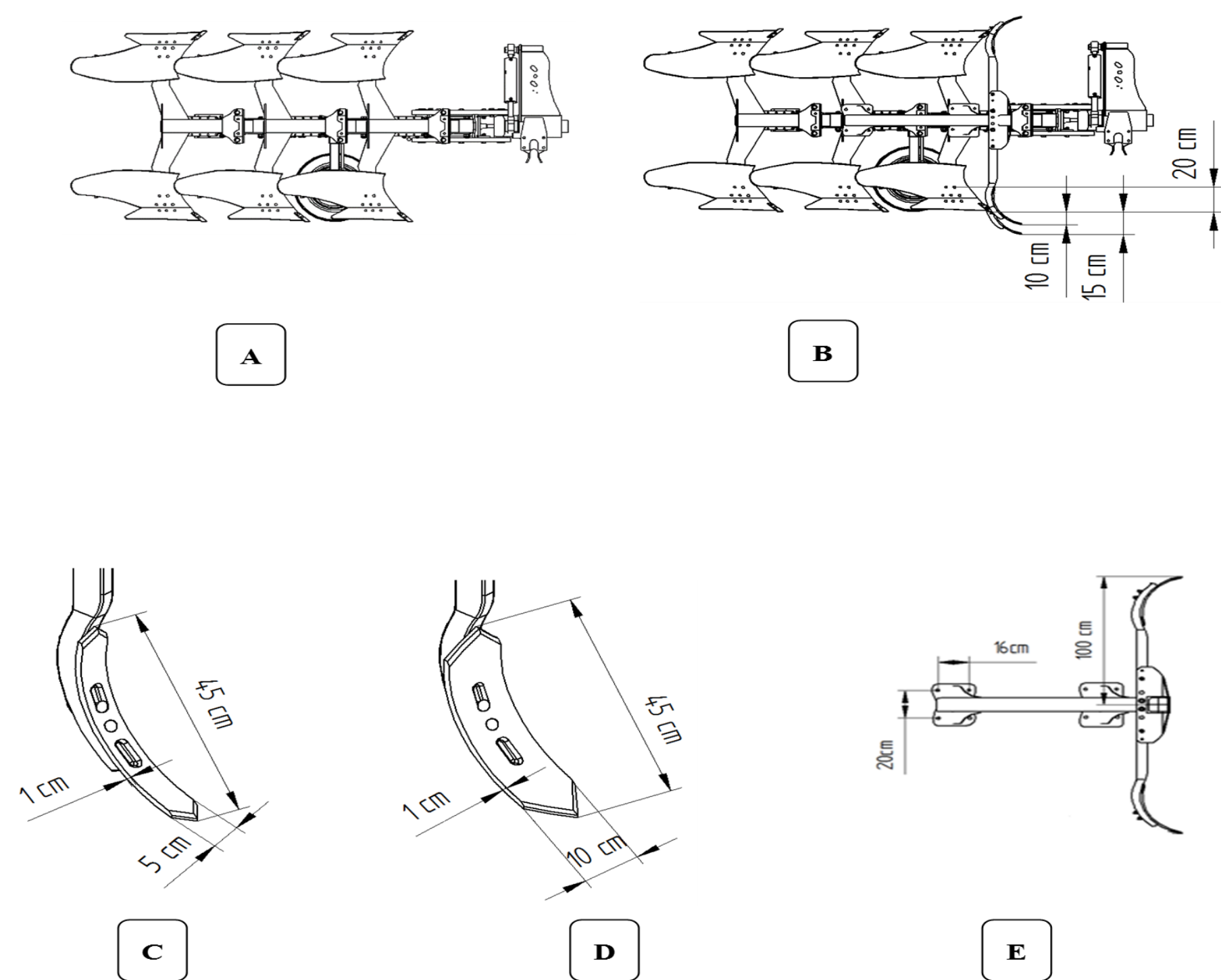


Figure 1. Schematic drawing of the reversible moldboard plough and extra tine: A – Normal plough without tine, B – modified plough, C – TINE1, D – TINE2, E – tines with shanks

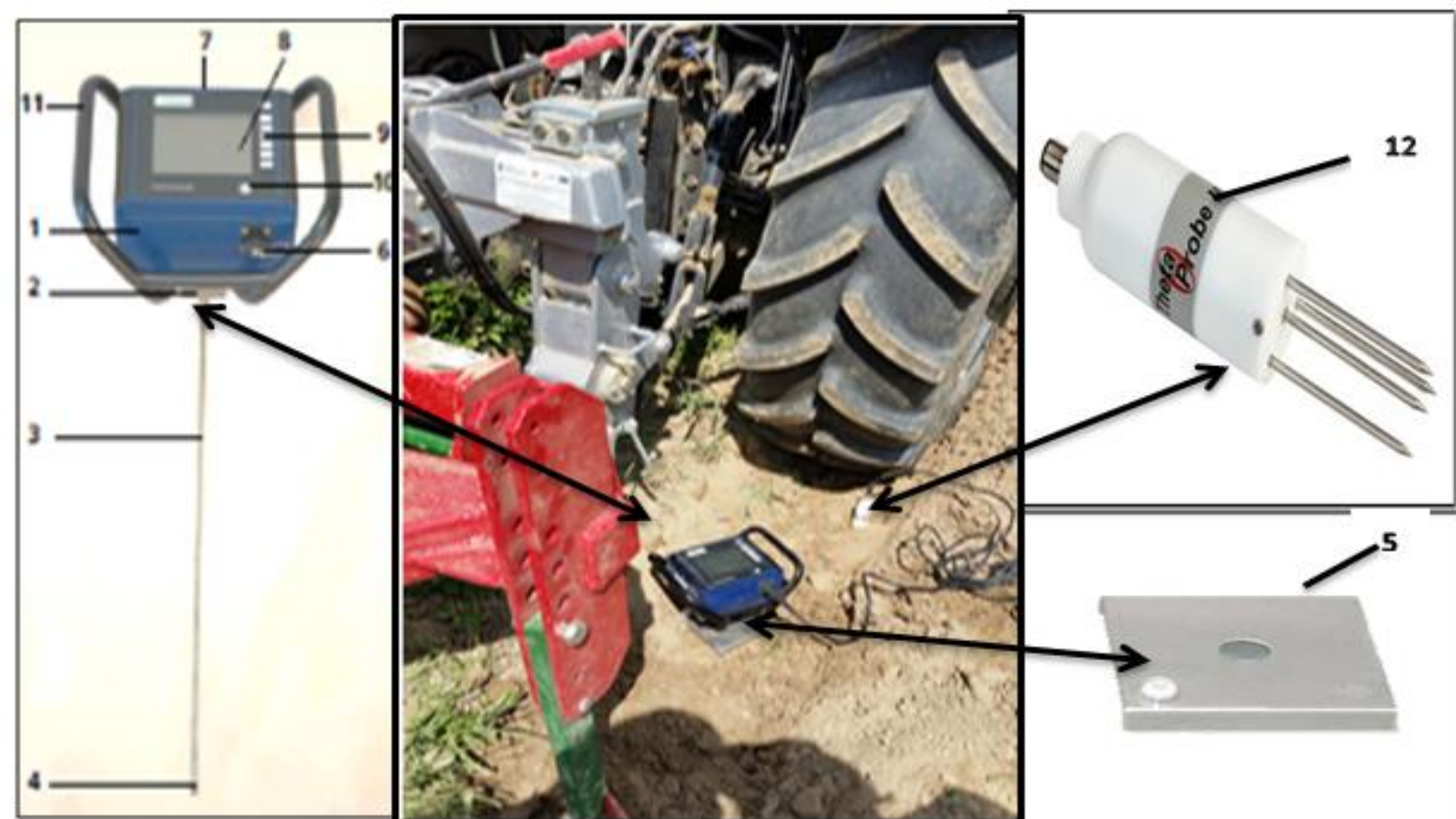


Figure 2. Eijkelkamp penetrolgger: 1– water–resistant housing, 2– impact absorber, 3– bipartite probing rod,

4 – cone, 5– depth reference plate, 6– communication port, 7– GPS antenna, 8– LCD screen, 9 – control panel, 10 – level, 11– electrically insulated grips, 12 – soil moisture measuring sensor (theta probe)



Figure 3. Three hitch dynamometer (frame)

4. Results

Figure 4 shows that the magnitude of the horizontal force for all tine settings (depth, width) operating below the moldboard plow were significantly higher than that of the NO TINE treatment, and it tends to be increasingly significant ($p \leq 0.05$) with tine depth and its width, this becomes pronounced when the tine depth was increased from 10 to 15 cm, the figure shows that TINE2D15 recorded the highest value of mean horizontal force (13 kN), due to the its higher working depth and width. Moreover, at the same working depth, the horizontal force was significantly increased by TINE2 in comparison with TINE1, this could be due to the effect of tine width. Although, TINE2D10 had twice the width of TINE1D15, a lower horizontal force was recorded, this indicates that the effect of tine width was less than the effect of its depth on the horizontal force

Figures 5 shows that at depth of 20-25 cm, an increase in penetration resistance was produced by the NO TINE treatment compared to the other soil depths,. This could be the result of a wheel pass on the bottom of the furrow during plowing operation, thus increased soil compaction, the effect decreased with soil depth, probably for two reasons, firstly, the effect of wheel traffic decreased with soil depth, secondly, soil depth of 25-30 cm has been previously compacted and exceeded the critical value of 2 MPa, therefore, it was less affected by the tire pass. The cone index was significantly decreased ($p \leq 0.05$) after the operation of the tine, within the hardpan layer, this was due to the fact that the soil has been tilled and pulverized which leads to less resistance to penetration into the soil.

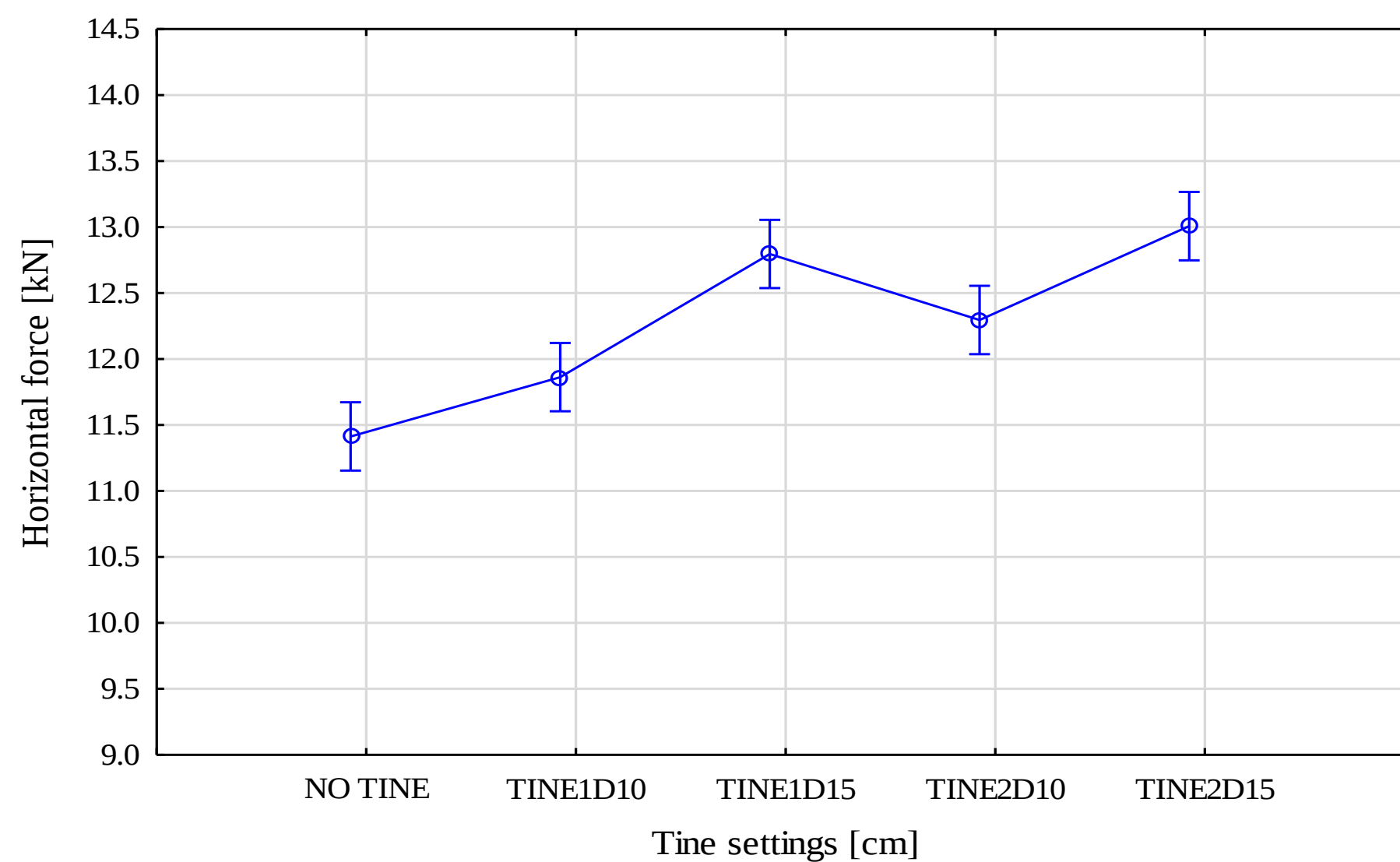


Figure 4. Effect of tine settings on the horizontal force [kN]. Moldboard plow without extra tine (NO TINE), plow with tine of a width of 5 cm (TINE1), plow with tine of a width of 10 cm (TINE2), working depth of tine (D). The different lowercase refers to the 1st location and the different uppercase refers to the 2nd location are significantly different according to Tukey test ($p \leq 0.05$)

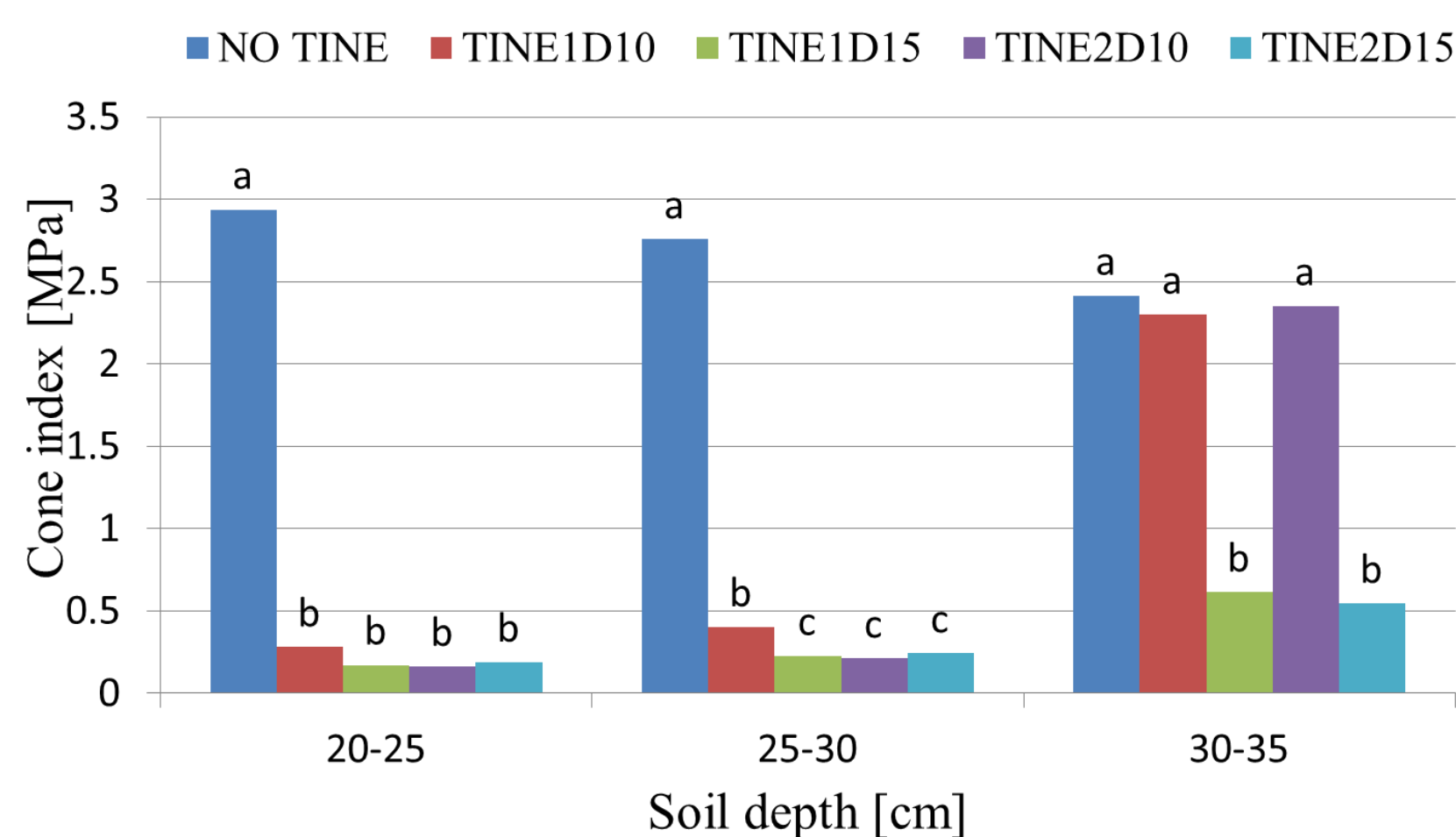


Figure 5. Effect of tine settings on the Cone index [MPa]. The moldboard plow without an extra tine (NO TINE), plow with tine at a width of 5 cm (TINE1), plow with tine at a width of 10 cm (TINE2), tine working depth (D). Means followed by the same letter are not significantly different according to the Tukey test ($p \leq 0.05$)

5. Conclusion

1. An increased in horizontal force requirement by modified plow in comparison to normal one, this increasing was higher affected by increasing extra tine depth than its width.

1-The wheel traffic on the bottom of the furrow during plowing operation using a plow without an extra tine produced a higher value of penetration resistance.

2- The modified plow resulted in a disturbed and stirred the hardpan layer below wheel pass in the furrow, this led to a reduction in the value of the penetration resistance